VOL·X NO·6

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

S·A·E

JUNE 1922

SUMMER MEETING NUMBER

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

THE

JOURNAL
OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

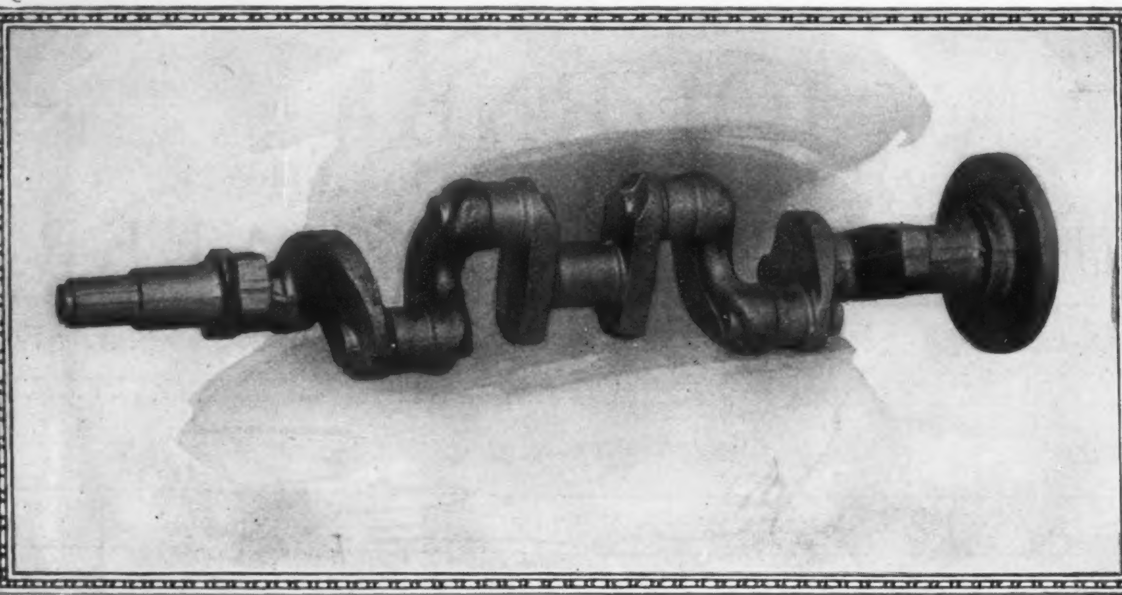
Published by the SOCIETY OF AUTOMOTIVE ENGINEERS, INC., *29 W. 39th St., New York*

Issued monthly, $1 a number, $10 a year; foreign $12 a year; to members, 50 cents a number, $5 a year. Entered as second-class matter, Aug. 23, 1917, at the post office at New York, N. Y., under the Act of Aug. 24, 1912. Acceptance for mailing at special rate of postage provided for in Section 1103, Act of Oct. 3, 1917, authorized on Aug. 2, 1918.

B. B. BACHMAN, *President* COKER F. CLARKSON, *Secretary* C. B. WHITTELSEY, *Treasurer*

Vol. X JUNE, 1922 No. 6

INDEX



THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. X June, 1922 No. 6

Chronicle and Comment

June Council Sessions

THE Council of the Society will hold a meeting at White Sulphur Springs on Monday, June 19. This will be a regular session. A special session will be convened on the following afternoon for the purpose of attending to unfinished or new business, including consideration of the action taken by the Standards Committee on Tuesday, June 20.

Part II 1920 Transactions

COPIES of the forthcoming part of TRANSACTIONS, Part II of Vol. 15, will be distributed by parcel post early this month to the members who have ordered them. The issue contains about 40 papers and reports, together with the discussion thereon, presented at the 1920 Summer Meeting of the Society and at various meetings of the Sections.

Hoover on S. A. E. Standards

THE S.A.E. HANDBOOK appears to be a very complete example of the sort of technical standardizations that would greatly decrease our industrial wastes if they could be properly applied to our industries. Through our Division of Simplified Practice we hope to aid in these applications and it seems that we can be of special service in the automobile field.—Herbert Hoover, Secretary of Commerce.

The Mechanism of Lubrication

THE paper by Robert E. Wilson and D. P. Barnard, 4th, presented at the Annual Meeting and to be printed in an early issue of THE JOURNAL, should be of unusual interest to all engineers who have made a study of the subject of lubrication. The authors give only a very brief review of the classical theory of lubrication, and follow this with a very illuminating and practical discussion of some recent developments.

Among other things, they state that the coefficient of friction of a plain bearing depends upon a definite relation between speed, viscosity and bearing pressure and not on any one of these independently. The simple expression for this relation, which is $f = k z N/p$, where f denotes the coefficient of friction, k is a factor of proportionality, z the coefficient of viscosity, N the speed and p the bearing pressure, can be made very useful in designing bearings and detecting their shortcomings.

The authors explain what is meant by oiliness of lubricants; show why nearly all friction-machine tests have omitted this factor, and promise a future paper covering new work from the Massachusetts Institute Laboratory on this subject.

1922 Roster

COPIES of the printed Roster of the Society as of April 1 were forwarded last month to members who had ordered them. This book of 358 pages, issued in paper-bound form in accordance with long-standing practice, is of course of considerable value to the members in their work, especially in connection with traveling and correspondence. It contains lists of members arranged alphabetically and geographically and a list of the companies with which the members are associated, as well as the names of the officers and committeemen of the Society and of its representatives in other organizations, and statistical information with regard to classification and residence of the members.

Current Standards Reports

TWELVE Divisions of the Standards Committee of the Society will submit recommendations on 33 different subjects at the meeting of the Society to be held this month. The various recommendations relate to engineering practices in the agricultural equipment, electrical equipment, internal-combustion engine, iron and steel, lubricants, non-ferrous metals, parts and fittings and passenger-car body fields. Most of the matters submitted for review involve new standards, but 10 revisions of and 6 extensions of present standards are recommended. The cancellation of one standard is to be considered; and one progress report will be made. The reports of the respective Divisions are printed in this

issue of THE JOURNAL and a comprehensive discussion of them at White Sulphur Springs is desired.

Society Nominating Committee

THE Nominating Committee of the Society consists of members elected by the Sections, each Section naming one man and an alternate, and three members of the Society elected at the business session of the Semi-Annual Meeting of the Society preceding the Annual Meeting at which the officers are elected. Obviously, the committee is charged with the conduct of work of fundamental importance to the Society. The following have been named for service on the 1922 Committee by the Sections:

Section	*Member*	*Alternate*
Buffalo	A. A. Gloetzner	F. W. Gurney
Cleveland	R. J. Nightingale	W. R. Strickland
Dayton	V. G. Apple	L. S. Keilholtz
Detroit	Howard A. Coffin	T. J. Lítle, Jr.
Indiana	Mark A. Smith	W. G. Wall
Metropolitan	Cornelius T. Myers	Finley R. Porter
Mid-West	B. S. Pfeiffer	Taliaferrc Milton
Minneapolis	L. A. Emerson	
New England	Leon W. Rosenthal	Prof. E. P. Warner
Pennsylvania	T. F. Cullen	F. M. Germane
Washington	W. S. James	

The additional members of the committee are to be elected at White Sulphur Springs on the evening of June 20. Under the rules of the Society the committee will hold an organization session during the Semi-Annual Meeting, and in due course nominate members to serve as officers of the Society during 1923.

Engineering Training for Foreign Service

THE demand for the training of engineers in foreign languages comes at a critical hour. Engineering colleges are hard pressed to meet the demand for the time devoted to civic and economic training. At many of the leading schools it has been keenly felt that engineering students as a rule did not, by the language training given, acquire the ability to read foreign technical literature, much less to express themselves in a foreign tongue.

American engineers should be able to read foreign literature and to deal with foreigners in their own tongue. In connection with work abroad the need often arises for knowledge of a language never taught in school. In European trade Russian, in Asiatic trade Chinese, in Brazilian trade Portugese are languages that undoubtedly will be of great importance in the future. Yet, we can hardly think of teaching them to engineering students generally.

The thing to do, therefore, seems to be to develop in our students the ability to acquire knowledge rapidly and unaided in a new field when the need of such knowledge arises. This is, however, exactly the ability that our engineering schools leave undeveloped. Foreign schools and our graduate schools develop it; the latter, however, at a too late age. By excessive coaching, supervising and scheduling we kill rather than foster initiative and enterprise in our undergraduates. This, to my mind, is the cardinal defect in our American college education. We develop perhaps the most moral, generous and decent type of man in the world. And this must not be given up. But we suppress the quality without which we will not hold our own in technical or commercial competition with foreign nations.

If any cure is to be wrought in this matter, it must come through missionary work of business and of the United States Government. The students must be made to see the opportunities offered them in foreign service. They must be given concrete examples of the unusual possibilities in new and undeveloped fields, where highly trained men are scarce. They must be imbued with an enthusiasm for real individual achievement and for far-reaching work. American engineering students are high-grade material, but they are now being trained to be routine men and job-holders at home rather than to be energizers and vitalizers on a world scale. They must be given vision and self-reliance. Let us hope that business and the Government will see their duty in this matter and set to work on a missionary campaign at an early date.—Prof. C. A. Norman at Congress on Commercial Engineering.

Highway Research

THE ability of concrete slabs to withstand repeated stresses caused by the passage of heavy trucks is being investigated at three laboratories, it is reported by Director W. K. Hatt, of the Advisory Board on Highway Research. Dean Johnson, of the University of Maryland, has devised a machine by which concrete beams may be deflected a few thousandths of an inch at the rate of 200 times per min. Fundamental research of this kind is not only of value in highway research but throws new light upon the fundamental properties of materials.

The composition of asphalt mixtures that will be stable at widely varying temperatures is being investigated by H. S. Mattimore, of Pennsylvania. The Bureau of Public Roads has built a circular track that will be surfaced with several mixtures of bituminous materials and subjected to truck action to determine the cause and remedy of the waving of pavements, which last summer involved enormous maintenance costs. Another circular track will expose various concrete aggregates to the wear of traffic.

Dr. G. E. Ladd, of the Bureau of Public Roads, has completed a report on his study of corrugations in gravel roads in eight different States. Contrary to the general impression of drivers, these waves, which have been so expensive in the maintenance of roads and of vehicles, are from 25 to 35 in. in length, crest to crest, and have a maximum height of $1\frac{1}{2}$ in. unless pitting and ravelling have begun. These corrugations, it is stated, arise from the action of motor vehicles. Generally in gravel roads they originate in the kick-back of surface materials arising from the spin of one or both of the rear wheels as they descend from a bounce over some obstacle or depression. Spring action has undoubtedly a contributing and modifying effect. Dr. Ladd determines the density of traffic at which such corrugations arise and a means of keeping them down.

The problem of the sand-clay or top-soil road, which must furnish the main transportation for large regions of the South Atlantic states, is being studied in Georgia and North Carolina. Prof. C. M. Strahan, of the University of Georgia, has developed data that serve to define the percentage of clay and coarse sand that will render these roads stable under a traffic that they can reasonably be expected to carry. The Bureau of Public Roads and the North Carolina Highway Commission also are investigating this type of road to determine the limit of traffic under which they can be maintained, and a means of improving and extending the life of the surface.

(*Concluded on page* 522)

The Summer Meeting

THERE is every reason to believe that the 1922 Summer Meeting at White Sulphur Springs will excel those of the past 2 or 3 years in point of attendance and general attractiveness. There can be no doubt that the meeting will be well attended, for the reservations have reached a total of 561 as this issue of THE JOURNAL goes to press. It is fortunate that two unusually fine hotels are available at White Sulphur Springs with sufficient capacity to house the members comfortably. Many pleasant rooms will be at the service of the late-comers but it is the safest plan to send in reservations as far in advance of the meeting as possible. A reservation blank was printed on page 70 of the May issue of THE JOURNAL and another copy was included with a recent issue of the *Meetings Bulletin*. All applications for reservations *mailed before June* 15 should be addressed to the Society office in New York City, but after that date they should be addressed to the Society's Headquarters, White Sulphur Springs, W. Va. The blanks must be accompanied by a check to cover the necessary reservation and guest fees.

General Plan of Meeting

The general meeting plan that has proved so successful in past summers will be followed in 1922. The mornings will be devoted to automotive engineering sessions at which valuable technical papers on problems of current interest will be presented and discussed. The afternoons are set aside for an elaborate program of sports and games, while the evenings will be spent watching the latest screen drama or one-stepping to the tune of a popular syncopated symphony. Dull moments will be non-existent at White Sulphur Springs, for the resort itself offers numerous attractive features to while away the fleeting hours, even though an elaborate program of entertainment were not provided.

First Day Devoted to Standards

The opening day of the Summer Meeting, Tuesday, June 20, will be the occasion of the semi-annual meeting of the Standards Committee. Morning and afternoon sessions will be held and the various Standards Committee Divisions will report their recommendations to the Committee for approval. The subjects to be considered cover a wide field and are worth careful study by those engaged in the manufacture of parts or materials affected by them. Objections to or constructive criticism of any of the proposals will be welcomed at this meeting. The complete group of reports is printed in this issue of THE JOURNAL. Following the voting on the standards reports, one or two brief addresses will be made on the value of engineering standards to industry. R. M. Hudson, of the United States Department of Commerce, has been selected by Secretary Herbert Hoover to tell something of the work that this government agency is fostering to reduce waste in industry through standardization.

B. B. Bachman, president of the Society, will address the members at the semi-annual Business Meeting, Tuesday evening. Reports of the standing committees, the Secretary and the Treasurer will be presented. The members-at-large of the Society Nominating Committee will be elected at this time to act with the Section delegates in the selection of a ticket of Society officers for the year 1923.

Motorbus and Aeronautic Sessions

Two technical sessions will be conducted simultaneously on Wednesday morning, June 21. They will be devoted to the Motorbus and Aeronautics respectively. G. A. Green will present a paper on motorbus design which will be based on the experience gained through his long association with this form of automotive transportation. Mr. Green's paper will deal principally with the factors affecting bus chassis and body design, indicating the major points of difference between motor truck and motorbus chassis construction. In his work with the Fifth Avenue Coach Co., Mr. Green has been fortunate in carrying the dual responsibilities of operator and designer. His paper, though written from the engineering viewpoint, reflects the practical experience of the operator on the street. The paper which R. E. Plimpton will read at the Motorbus Session will set forth accepted bus design practice as evidenced by the models being currently built by motorbus manufacturers. Mr. Plimpton will make an analysis of the types of service the motorbus is expected to fill and indicate the designs best suited to each. Because of the rapid expansion of the motorbus arm of transportation, it is natural to expect

an active and valuable discussion after the presentation of these two papers.

Prof. E. P. Warner, of the Massachusetts Institute of Technology, has contributed a paper on the mathematical prediction of airplane performance which he will read at the Aeronautic Session. Professor Warner has derived simplified formulas that will serve the airplane designer in estimating such performance factors as minimum and maximum speed, rate of climb and the ceiling of a plane. Values obtained from these formulas were checked by comparison with those of a large number of airplanes the performance of which was known. Application of the formulas to specific examples will be made. The paper is printed in this issue of THE JOURNAL. Capt. George E. A. Hallett, of the engineering division of the Air Service, will outline the general procedure of designing and developing airplane engines followed by the organization that he heads at McCook Field. It is Captain Hallett's belief that many of these methods are applicable to the experimental development of motor-car and motor-truck engines to reduce the time needed for experimental tests and insure an earlier arrival at the stage where designs are perfected and ready for production. He will supplement the paper, which is printed on another page, with a description of the progress in airplane engineering that has been made by the Air Service Engineering Division in the past year. He will illustrate this talk with many interesting slides. An analogy by C. L. Egtvedt, printed in this issue of THE JOURNAL, will be read by title and discussed. At the conclusion of the discussion there will be an exhibition of authentic aviation pictures taken during the war with Germany. These films have been loaned by the Air Service and have been shown rarely in public. They possess great historical interest, as well as emphasize the airplane's military importance and the need of our keeping pace with foreign aeronautical development.

THE RESEARCH SESSION

The Research Session will be held Thursday morning, June 22. The major topic of discussion will be the desired volatility of motor gasoline. Papers will be given by W. S. James and C. T. Coleman, of the Bureau of Standards, and Dr. H. C. Dickinson, manager of the Research Department of the Society. Mr. James' paper will describe an instrument, developed by the Bureau of Standards, that photographically records the performance data of an engine when it is propelling a vehicle on the road under typical driving conditions. Acceleration, rated fuel-consumption, road and relative air-speeds, engine and transmission temperatures and other factors are all reproduced for later analysis. Mr. James will include in his presentation the conclusions drawn from extensive road-tests observed and studied with this ingenious instrument. One of the studies was made to determine the effect of decreased and increased gasoline volatility upon car performance and fuel economy. Practically all of Mr. James' work has been conducted on passenger cars. C. T. Coleman's paper, on the other hand, relates principally to the changes in performance and economy of motor trucks resulting from a lowering or raising of the end-point of the gasoline used. The tests to be described by Mr. Coleman are now being made by the Bureau of Standards, using the entire fleet of Post-Office transportation units in Pittsburgh and Philadelphia. Two different road conditions are represented in these cities; Philadelphia streets are smooth with few hills; Pittsburgh streets are rough with many hills. Four fuels of varying volatility are being employed in the tests, these ranging from aviation gasoline to a very heavy grade of gasoline. Fuel-consumption, oil dilution and oil consumption will be carefully observed and the resultant statistical data analyzed. Mr. Coleman will present the conclusions of this analysis in his paper at the Research Session.

Following these two papers, Dr. H. C. Dickinson will present the progress report of the Research Committee of the Society. He will give a complete and detailed description of the Society's Fuel Volatility Tests that are being conducted by the Bureau of Standards, Bureau of Mines and many of the motor-car companies under the direction of the Research Department of the Society. These tests are being financed by the American Petroleum Institute, the National Automobile Chamber of Commerce, and assistance is being given by the participating oil refiners and motor-car builders. The object of the tests is to determine what sort of petroleum fuel, as regards volatility, will give the average vehicle operator the most ton-miles of transportation per barrel of crude oil used in producing the fuel. Dr. Dickinson will call upon some of the chief engineers of motor-car companies participating in the tests for brief outlines of the procedure being followed in the work under their direction. It is only natural to anticipate the deepest interest in the Research Session. The knowledge gained in this one technical session should fully justify the expense of attending the Summer Meeting.

PASSENGER CAR SESSION

The technical session to be held Friday morning, June 23, has been designated as the Passenger Car Session. H. M. Crane will present his paper on A New System of Spring Suspension for Automotive Vehicles, which is printed in this issue of THE JOURNAL. He will discuss those factors influencing proper rear-axle spring suspension that have led him to design the rather unusual suspension described in the paper. It is the general belief among automobile critics that spring suspension and riding quality are features of present-day motor cars that warrant closer study on the part of the automotive engineer. For this reason it is hoped that Mr. Crane's paper will arouse an active discussion.

Engines with overhead camshafts and valves are becoming more popular in Europe and it is possible that a greater demand may be created for similar engines in this Country. P. M. Heldt's paper calls attention to the claims made for the overhead-camshaft engine and presents the mechanical features of a number of the popular European examples. The paper is printed in this issue of THE JOURNAL. The overhead-camshaft engine has been used in airplanes, motorboats and racing cars and has many advantages to recommend it in these instances. There is considerable difference of opinion, however, about its being the most practical type for the average operator of passenger cars. F. A. Bonham will read a paper presenting some pertinent suggestions on what the automotive engineer can do to assist the service and parts departments. Mr. Bonham has a number of constructive ideas that warrant attention and thorough discussion.

FUEL AND ENGINE SESSION

The Fuel and Engine Session on Saturday, June 24, will conclude the Summer Meeting. It will include four papers that contribute a large amount of valuable information for the engineer engaged particularly in the design of automotive powerplants. Thomas Midgley, Jr., will present the quantitative results of an analysis of the

Summer Meeting Program

TUESDAY, JUNE 20	
10 a. m. and 2 p. m.	STANDARDS SESSION Standards Committee Reports and Discussion Papers on Standardization
2 p. m.	TENNIS AND GOLF TOURNAMENTS START
8 p. m.	SEMI-ANNUAL BUSINESS MEETING Address of President Reports of Committees and Officers Election of Nominating Committee Members
WEDNESDAY, JUNE 21	
10 a. m.	MOTORBUS SESSION G. A. GREEN—The Design of the Motorbus R. E. PLIMPTON—Characteristics of Present-Day Motorbuses
10 a. m.	AERONAUTIC SESSION E. P. WARNER—Airplane Performance Formulas G. E. A. HALLETT—A Method of Developing Aircraft Engines C. L. EGTVEDT—Flighty Reflections Air Service Motion Pictures of the War with Germany
2 p. m.	Men's Golf Qualifying Round Men's Tennis—Singles Inter-Section Baseball
5 p. m.	Orchestra Concert
8 p. m.	AQUATIC SPORT PROGRAM
9:30 p. m.	Motion Pictures Dancing
THURSDAY, JUNE 22	
10 a. m.	RESEARCH SESSION H. C. DICKINSON—Report of the Activities of the Research Department of the Society. Information Service Fuel Research Highway Research W. S. JAMES—Fuel Volatility Research at the Bureau of Standards, with Demonstration of Test Car Equipment. C. T. COLEMAN—Report of the Bureau of Standards Fuel Volatility Tests on Motor Trucks. Report of Chief Engineers of Companies Cooperating in Fuel Volatility Tests.
2 p. m.	ANNUAL S. A. E. FIELD DAY Track and Field Events Men's Golf Match Play Men's Tennis—Doubles and Singles
4 p. m.	Inter-Section Baseball
5 p. m.	Orchestra Concert
8 p. m.	Section Stunts
9:30 p. m.	Motion Pictures Dancing
FRIDAY, JUNE 23	
10 a. m.	PASSENGER CAR SESSION P. M. HELDT—Overhead Camshaft Passenger-Car Engines H. M. CRANE—New System of Spring Suspension for Automotive Vehicles F. A. BONHAM—The Automotive Engineer and the Service Problem
2 p. m.	FINALS IN ALL SPORTS EVENTS AND TOURNAMENTS
5 p. m.	Orchestra Concert
8 p. m.	GRAND BALL
9:30 p. m.	Motion Pictures Dancing Contest Presentation of Prizes
11 p. m.	Buffet Supper
SATURDAY, JUNE 24	
10 a. m.	FUEL AND ENGINE SESSION F. C. MOCK and M. E. CHANDLER—The Hot-Spot Method of Heavy-Fuel Preparation. THOMAS MIDGLEY, JR., and T. A. BOYD—Detonation Characteristics of Some Blended Motor-Fuels. G. A. ROUND—Oil-Pumping A. A. BULL—Oil Consumption

Note: An appropriate program of entertainment, sports and cards will be conducted for the ladies each morning during the Technical Sessions.

Sports Events

Golf—Tuesday through Friday

Men's Championship Tournament
Men's Flag Consolation
Men's Driving Contest
Men's Putting Contest
Ladies' Championship Tournament
*Ladies' Putting Contest

Tennis—Tuesday through Friday

Men's Singles Tournament
Men's Doubles Tournament
*Ladies' Singles Tournament
Mixed Doubles Tournament

Track—Thursday Afternoon

*50-yd. Dash—Men under 30
50-yd. Dash—Men 30 to 40
50-yd. Dash—Men over 40
*50-yd. Dash—Boys under 12
*100-yd. Dash—Boys under 16
*50-yd. Dash—Ladies
*50-yd. Dash—Girls under 16
Fat Man's Race—Men weighing more than 3 lb. per in. of height
Three-Legged Race—Men
Potato Race—Men and Ladies
One-Legged Race—Hopping with legs tied

Field—Thursday Afternoon

Shot Put—Men
Hop, Step and Jump—Men
Standing Broad Jump—Men
*High Jump—Men
*Throwing Baseball—Ladies
*Egg Race—Ladies
Inter-Section Relay — Teams of four men

Baseball—Tuesday through Friday

For S.A.E. Section Championship Cup —Teams of nine—indoor or soft ball used

Trapshooting—Tuesday through Friday

For S.A.E. Section Champion Medal (Bring your own gun.)

Swimming—Wednesday Evening

33-yd. Swim—Men
33-yd. Swim—Ladies
*66-yd. Swim—Men
*66-yd. Swim—Ladies
Plunge for distance—Men
Plunge for distance—Ladies
*Fancy Diving—Men and Ladies
Inter-Section Relay — Teams of four men
Several Stunt Races

*Only these events open to guests

properties of blended motor-fuels. The data relate particularly to the comparative detonation values of the several fuels as determined by the bouncing-pin method. Mr. Midgley's paper appears in this issue of THE JOURNAL. Although much study has been made and many papers written on the subject of hot-spot manifolds, F. C. Mock and M. E. Chandler find a general misconception of the engineering fundamentals that must be followed in designing efficient manifolds of this type. Their paper collates many of the data now in existence and adds to them some pertinent information resulting from tests just completed by the authors.

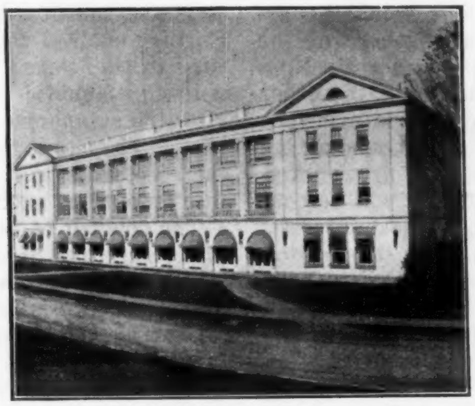
One of the strongest and most persistent complaints received through the average service department is that of engines pumping oil. The trouble is aggravated by lowered fuel volatility, poor carburetion, and crankcase-oil dilution. Doubtless many of the members have spent much time and effort in seeking a means of preventing or alleviating this difficulty. Excessive carbon-deposit and fouled spark-plugs are the most common complaints chargeable to this cause. Appreciating the urgency of the problem, the Meetings Committee has secured two papers on the subject of Oil-Pumping which will be given at this Session. A. A. Bull has prepared a searching analysis of the causes of oil-pumping and excessive oil consumption in his paper. G. A. Round has approached the matter independently and from the lubricating engineer's viewpoint. Both of these papers are printed in this number of THE JOURNAL and are deserving of close study so that they may be thoroughly discussed at the time of their presentation at White Sulphur Springs.

Written Discussion of Papers

Eight of the papers to be read at the Summer Meeting are printed in this issue of THE JOURNAL. Preprints or abstracts of the other papers on the program will be sent to those who order them on the blank recently mailed to all members. Publication of the papers in advance of the meeting enables the members to study these important automotive engineering contributions with care. It provides an opportunity for the preparation of written discussion. Such written discussion need not be withheld until the time of the Summer Meeting. It should be forwarded to the Society offices where arrangements will be made for its presentation in the proper technical session. This suggestion applies particularly to those unable to attend the meeting and present their comments orally. It often happens that the information brought out in the discussion is of equal or greater value than that presented by the author in his paper. Read the Summer Meeting papers that touch upon matters in your field. If your experience is contradictory to that of an author, by all means submit it. If it supplements his opinions, strengthen his statements by contributing your own knowledge of the problem treated. Written discussion is a highly desirable addition to the papers, and members are strongly urged to mail such discussion to the Society offices in the case of the 1922 Summer Meeting papers.

Railroad Transportation

The reduced-fare privilege that has been secured from the railroads differs from the one in effect at the Annual Meeting last January. The members will receive a reduced-fare certificate about June 10. When transportation to White Sulphur Springs is purchased, this certificate entitles the member to a reduction of 25 per cent on the cost of a round-trip ticket. These certificates cannot be used by non-members, but they are valid for tickets purchased for dependent members of a Society member's


family. Round-trip tickets only may be purchased under this plan, no stop-overs are allowed and the same route must be used in both the trip to White Sulphur Springs and return. The Society has been fortunate in securing this reduced-fare concession from all of the railroad associations in the United States and the consequent saving to the members will be a substantial one. The rates from the principal automotive centers are shown in the accompanying table which is reprinted from the May issue of THE JOURNAL with one or two corrections.

Special Trains and Pullmans

Before making any Pullman reservations, get in touch with the Secretary of the Section nearest you and arrange to travel to the meeting in special cars with your fellow members. Two special trains have been scheduled to transport the Eastern and Western members to White Sulphur Springs. The Detroit, Cleveland and Midwest Sections have reserved special Pullman cars leaving their respective cities on Monday afternoon, June 19, and arriving in Cincinnati on the evening of the same day. Here they will be made-up into an S.A.E. Special all-Pullman train which will arrive in White Sulphur Springs, Tuesday morning, June 20, soon after breakfast. The members in northern Ohio can board the Cleveland cars; the members in Toledo and Michigan can be accommodated on the Detroit cars; the Indiana members should make reservations on the cars from Chicago.

The special train from the East will carry Pullmans from New York City and Philadelphia. New England and New York State members can make reservations on this train through the Society offices. The Eastern train leaves New York at 3:40 p. m., Eastern Time, on Monday afternoon, June 19, and arrives in White Sulphur Springs Tuesday morning before breakfast. Washington members can board this train, as it passes through their city.

The complete time-schedule of the special trains is given in the accompanying table, with the corresponding railroad fares. The Meetings Committee hopes to be able to announce soon the arrangement for similar special trains returning on Saturday, June 24, after the completion of the meeting. These trains will enable the members to remain at White Sulphur Springs until after lunch Saturday and still reach their home cities on Sunday morning, June 25.

It is very important that all members taking advantage of the special reduced-fares have their return-trip tickets validated by the railroad agent at White Sulphur Springs before boarding the return train. This portion of the fare-and-half ticket is not valid until the stamp of the White Sulphur Springs agent appears on it.

A Very Complete Sports Program

With the unusually complete sport facilities provided at White Sulphur Springs, it was to be expected that a

RAILROAD ACCOMMODATIONS TO WHITE SULPHUR SPRINGS, W. VA., FOR SUMMER MEETING JUNE 20-24, 1922

		TIME SCHEDULE				RAILROAD FARES	
City	Railroad	Leave Monday, June 19	Arrive Cincinnati	S. A. E. Special Leaves Cincinnati	Arrive at White Sulphur Springs	Fare-and-Half Round-Trip	Tourist Round-Trip
Cincinnati	Chesap'ke & Ohio			**9:45 p.m.**	8:55 a.m.	$19.08	$20.62
Detroit	Big Four	11:45 a.m.	**8:55 p.m.**	**9:45 p.m.**	8:55 a.m.	28.08	31.97
Toledo	Big Four	**1:35 p.m.**	**8:55 p.m.**	**9:45 p.m.**	8:55 a.m.	24.98	28.22
Chicago	Big Four	**1:00 p.m.**	**9:00 p.m.**	**9:45 p.m.**	8:55 a.m.	34.47	38.70
Indianapolis	Big Four	**6:15 p.m.**	**9:00 p.m.**	**9:45 p.m.**	8:55 a.m.	25.01	27.75
Cleveland	Big Four	**3:00 p.m.**	**9:35 p.m.**	**9:45 p.m.**	8:55 a.m.	29.01	31.11
Dayton	Big Four	**7:57 p.m.**	**9:35 p.m.**	**9:45 p.m.**	8:55 a.m.	22.02	22.73

City	Railroad	Leave Monday, June 19	Arrive Washington	Leave Washington	Arrive at White Sulphur Springs	Fare-and-Half Round-Trip	Tourist Round-Trip
Boston	New Haven	8:00 a.m.	**7:40 p.m.**	**10:15 p.m.**	7:05 a.m.	$37.76	$49.04
Providence	New Haven	10:06 a.m.	**7:40 p.m.**	**10:15 p.m.**	7:05 a.m.	36.74	45.88
New Haven	New Haven	11:54 a.m.	**7:40 p.m.**	**10:15 p.m.**	7:05 a.m.	29.28	37.74
New York City	Pennsylvania	**3:40 p.m.**	**9:25 p.m.**	**10:15 p.m.**	7:05 a.m.	25.37	30.92
Manhattan Transf.	Pennsylvania	**3:58 p.m.**	**9:25 p.m.**	**10:15 p.m.**	7:05 a.m.	25.37	30.92
West Philadelphia	Pennsylvania	**5:56 p.m.**	**9:25 p.m.**	**10:15 p.m.**	7:05 a.m.	20.51	24.89
Baltimore	Pennsylvania	**8:30 p.m.**	**9.25 p.m.**	**10:15 a.m.**	7:05 a.m.	15.32	18.68
Buffalo	Pennsylvania	9:10 a.m.	**8:50 p.m.**	**10:15 p.m.**	7:05 a.m.	34.25	39.56
Washington	Chesap'ke & Ohio			**10:15 p.m.**	7:05 a.m.	13.16	17.28

very fine program of athletic diversions would be arranged. There will be much of the atmosphere of the intercollegiate meet in evidence at this Summer Meeting. Medals and cups are to take their place with the other handsome prizes of former years. Appropriate medals will be awarded to the golf, tennis and trapshooting champions. A prize goes to the member who proves himself to be the best all-round athlete. A cup will be contested for by the Section baseball teams. Most interesting of all, however, will be the struggle for the Section athletic championship. Predetermined point-scores have been assigned to the first, second and third places in all athletic events, including the baseball, golf and tennis tournaments. These scores will be credited to the Section represented by the victorious members or teams. When all sports events are concluded, the scores will be totaled and the championship cup awarded to the Section scoring the greatest number of points. In cases where a member is not affiliated with a Section, his points will be credited to the Section nearest him geographically as determined beforehand by the Sports Committee.

The golf tournament will be opened with a qualifying round, after which the contestants will be grouped into two or three flights based on their qualifying scores. The championship medal will be awarded to the winner in the first flight. The other flight winners will receive prizes in each case but these places will not contribute points to the Section Championship score.

The excellent tile swimming-pool at White Sulphur Springs enabled the Sports Committee to add a new feature to this year's program. On Wednesday evening a very fine aquatic program will be conducted. All may view the submarine dexterity of our more versatile members. The stunt races will be particularly amusing to the spectators and the sprinting and diving will create a fair amount of excitement.

The conduct of the sports program will be in the hands of a committee of which Mason P. Rumney is chairman. Mr. Rumney has announced the following as the personnel of his committee, each member being responsible for the branch of sports opposite his name.

SPORTS COMMITTEE

Mason P. Rumney, *Chairman*

Golf	Frank Lawrence
Tennis	Robert K. Floyd
Swimming	C. H. Brennan, C. B. Waterman
Baseball	Neil McMillan, Jr., E. R. Stroh
Field and Track	Lloyd P. Jones, E. V. Rippingille
Inter-Section Contest	Walter C. Keys
Clerk of Course	C. H. Van Sicklen
Professional Director	Victor Tomlinson

The complete list of sports events and tournaments is printed in another column. Entries should be made at the headquarters of the Sports Committee upon arrival at White Sulphur Springs.

GERMAN EXPORT TRADE

THERE is reason to believe that the advantage to the German export trade as a whole has been exaggerated. Certainly the available figures of aggregate German exports do not yet bear out the fear that Germany is making extraordinary progress in the world's markets. Moreover, as the German wage-earners and the general public have become more and more accustomed to think in terms of the external value of the mark, it appears that operating costs have tended to advance more nearly in proportion to advances in general prices. Again, the advantage obtained by the exporter who, for example, has been able to sell his goods for dollars and to buy marks at reduced external values, is in part offset by the price that he must pay for the imported raw materials necessary to the manufacture of goods for later exportation. In many other ways the apparent initial advantage is in part offset by corresponding disadvantages.

In specific lines of industry, however, there is no doubt that the German exporter has enjoyed great initial advantages. Manufacturers of other countries, in given cases, have actually faced cut-throat competition from Germany that for the time has seemed to threaten the existence of part of their foreign trade. Moreover, goods sold by Germany in the domestic markets of other nations have actually threatened to destroy the home market for some of the manufacturers of those nations.

While in the case of individual industries the German competition has been a real menace, the most important obstruction to normal trade recovery, so far as the German problem is concerned, is not to be found in the actual amount of destructive competition. On the contrary it is to be found in three other facts; the general uncertainty thrown into the entire trade situation by threatened, rather than actual, German competition; the danger, incurred in making contracts for future delivery, arising from the wide fluctuations of the exchanges, and the stimulation to, and often the necessity of, diverting money and energy from productive effort in normal lines to speculative or financial transactions in extraordinary lines.—*Commerce Monthly.*

STANDARDIZATION

STANDARDIZATION carried out on a sound engineering basis has the following significant aspects:

- It enables the buyer and the seller to speak the same language, and makes it possible to compel competitive sellers to do likewise
- Better quality of the product through ability of the manufacturer to concentrate on better design and through the reduction of manufacturing expense
- It lowers the unit cost to the public by making mass production possible
- By simplifying the carrying of stocks, it makes deliveries quicker and prices lower
- It decreases litigation and other factors tending to disorganize industry, the burden of which ultimately falls upon the public
- It eliminates indecision in both production and utilization, a prolific cause of inefficiency and waste
- It stabilizes production and employment, by broadening the possible market, and by making it safer for the manufacturer to accumulate stock during periods of slack orders to an extent that would not be safe with an unstandardized product
- By focusing on essentials, it decreases the selling expense, one of the serious problems of our economic system
- By concentrating on fewer lines, it enables more thought and energy to be put into designs, so that they will be more efficient and economical—Bulletin of the Chamber of Commerce of the United States.

Detonation Characteristics of Some Blended Motor-Fuels

By THOMAS MIDGLEY, JR.[1] AND T. A. BOYD[2]

SEMI-ANNUAL MEETING PAPER *Illustrated with* CHARTS

THE effects of admixtures of various percentages of alcohol and alcohol-benzene mixtures for reducing the detonating tendency of paraffin hydrocarbons have been measured by the authors. These results represent an extension of previous work in which similar determinations were made for benzene and other aromatic hydrocarbons. The bouncing-pin apparatus was used for making the determinations. The data obtained by its use are considered to have a high degree of accuracy.

In order that the effects of the blending materials might be measured through as wide a range as practicable, they were blended with kerosene for making the majority of the determinations. This made it possible to ascertain the characteristics of the materials up to a concentration of 80 per cent of benzene or 50 per cent of alcohol without introducing the difficulties due to excessively high engine compression. Because xylidine has the property of exerting a powerful suppressing action on detonation when present in a fuel in percentages that are relatively very small, the standard used as a basis of comparison in the tests was composed of small percentages of xylidine in the paraffin fuel. Tables and curves are appended that show the results of the tests in detail.

THAT the addition of benzene and other aromatic hydrocarbons to paraffin-base gasolines greatly reduces the tendency of these fuels to detonate when used in automobile engines has been known for some time. Also, it is well known that alcohol when blended with a paraffin-base gasoline improves the combustion characteristics of the fuel. The extension of the by-product coking industry in the United States during recent years has resulted in such an increase in the production of light oil that it can be absorbed only by the use of a part of the material as a motor fuel. This, coupled with the freedom from detonation that characterizes the combustion of benzol in engines, is causing the use of benzol-gasoline blends as motor fuels to be extended as rapidly as the benzol that is available will permit. The scarcity and high price of gasoline in countries where sugar is produced and the abundance of raw material for making alcohol there have resulted in a rather extensive use of alcohol for motor fuel in these districts. As the reserves of petroleum in this Country become more and more depleted the use of benzol, and particularly of alcohol, in commercial motor-fuels will probably become greatly extended.

Alcohol as produced commercially dissolves in gasoline only to a very small extent; but the addition of a proper percentage of an aromatic hydrocarbon, such as benzol, toluol or xylol, to the mixture renders the ingredients completely miscible. The availability of benzol for blending with motor fuels makes possible the use of alcohol in such mixtures. Blended motor-fuels containing the three ingredients, alcohol, benzol and gasoline have been sold in some parts of this Country for some time.

The object of this paper is to report the progress that has been made in measuring the detonating tendencies of mixtures of some of the principal materials that are used as components of the blended motor-fuels now available commercially. The detonation characteristics of aromatic hydrocarbons have been presented in a paper entitled Detonation Characteristics of Blends of Aromatic and Paraffin Hydrocarbons that is soon to be published in the *Journal of Industrial and Engineering Chemistry*. These results have since been extended by determining the detonation characteristics of blends of alcohol and of alcohol-benzol mixtures with paraffin hydrocarbons. Although these data are incomplete and have not been obtained in such a form as to be universally usable, it was thought advisable to present some of them in this way, especially in view of the fact that a considerable amount of the matter dealing with this subject that has been published in the past is in error. As examples of this the following statements by Ricardo may be cited:

(1) Xylene is inferior to toluene for the suppression of detonation[3]

(2) The detonation point of mixtures of ethyl alcohol, acetone, toluene and xylene with paraffin and other hydrocarbons, follows a straight law when the mixture is apportioned by weight and not by volume; that is, the addition of 40 per cent by weight of, say, toluene to hexane would raise the detonation point exactly four times as much as the addition of 10 per cent[4]

The primary reason for the unreliability of some of the data on blending characteristics that have been obtained and published is the previous lack of a means for measuring the detonating tendencies of fuels with sufficient accuracy. The use of the bouncing-pin method for the measurement of the intensity of detonation, however, gives results that are reliable and have a high degree of accuracy. This instrumentation, which is illustrated in Fig. 1 and was described in our paper entitled Methods of Measuring Detonation in Engines[5] that was presented at the 1922 Annual Meeting of the Society, has made it possible to secure the data that are presented in this paper.

In order that the effects of blending materials might be measured in as wide a range of concentrations as practicable, they were blended with kerosene for making the majority of the determinations reported in this paper. The greater tendency of kerosene than lighter paraffin hydrocarbons to detonate made it possible to determine

[1] M.S.A.E.—Chief of the fuel section, General Motors Research Corporation, Dayton, Ohio.

[2] Assistant chief of the fuel section, General Motors Research Corporation, Dayton, Ohio.

[3] See *Automobile Engineer* (London) March, 1921, p. 96; also THE JOURNAL, May, 1922, p. 311.

[4] See *Automobile Engineer* (London) March, 1921, p. 94.

[5] See THE JOURNAL, January, 1922, p. 7.

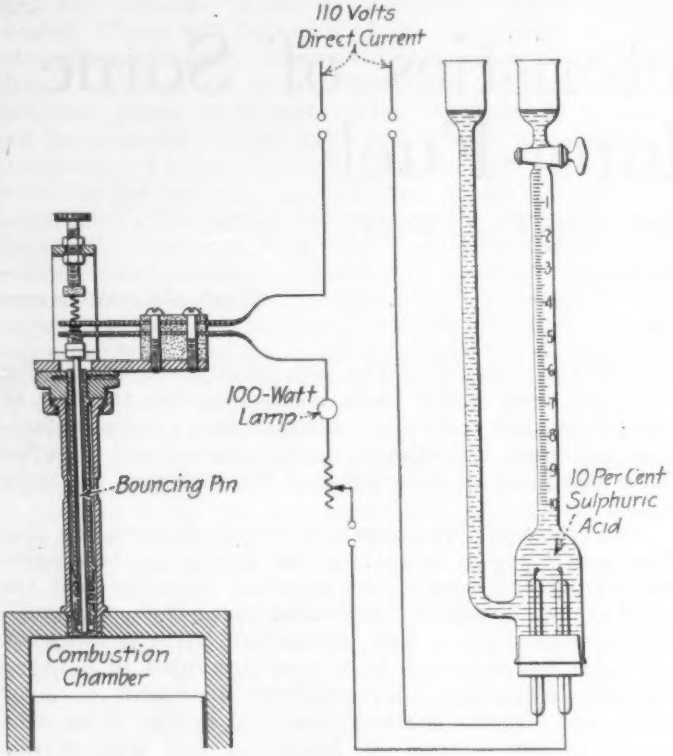


FIG. 1—ARRANGEMENT OF APPARATUS FOR MEASURING DETONATION BY THE BOUNCING-PIN METHOD

the detonation characteristics of blends up to a concentration of 80 per cent benzene or 50 per cent alcohol without introducing the difficulties incident to excessively high engine compression. The curves of Fig. 2 show that in general the characteristics of gasoline blends follow closely those of kerosene. This agreement is still better on the molecular basis, as is brought out in the previously mentioned paper on Detonation Characteristics of Blends of Aromatic and Paraffin Hydrocarbons. So that results obtained from a given concentration of blending material in kerosene are applicable within fairly close limits to blends of similar compositions in which the kerosene has been replaced by a gasoline.

On account of variations in engine conditions it is evident that data obtained from any particular engine are applicable in a quantitative way only to that one design and set of conditions. But, although widely different behavior may characterize the combustion of a certain fuel in two different engines, the relative behaviors of two given fuels will be comparative in whatever type of engine they may be run. Hence, in measuring the detonating tendency of any fuel it is essential that some standard be used as a basis of comparison. In the tests reported herein small percentages of xylidine in the same paraffin fuel that was used for blending with the alcohol and with the aromatic hydrocarbons were employed as a standard. Xylidine has the property, common to aromatic amines and considerably more marked in a number of other materials, of exerting a powerful suppressing action on detonation, when present in a fuel in percentages that are relatively very small. Thus, it may be seen from Fig. 2 that 1 per cent of xylidine in kerosene is equivalent for the elimination of detonation to about 15 per cent of benzene in the same material. This property of xylidine makes it possible to convert kerosene into a fuel that will withstand very high compressions without knocking, and with the addition of such a small percentage of xylidine that the combustion characteristics of the kerosene, other than its tendency to detonate, are not materially changed.

PROPERTIES OF MATERIALS USED AS FUELS

The materials used as ingredients of the various fuels that were either examined or employed as standards in the examinations, the results of which are reported in this paper and in the previous one referred to above, were "high-test" gasoline, commercial gasoline, kerosene, xylidine, benzene, or 90-deg. benzol, toluene, xylene and alcohol. The xylidine employed was a commercial material composed of the mixed xylidines. The alcohol used was absolute ethyl. The use of absolute rather than commercial denatured alcohol in these tests was necessary on account of the almost complete insolubility of the latter in paraffin oils, unless a "binder" such as benzol is used. Some physical properties of the other materials included in this list are presented in Table 1.

A 3/4-kw. Delco-Light engine was used for making all the determinations. This is a single-cylinder, air-cooled engine, direct-connected to a 32-volt, direct-current generator, and having a 2 1/2-in. bore and a 5-in. stroke. The engine was standard, except that a means was provided for adjusting the spark-timing, and that the compression was increased by stages from the normal ratio of 3.47 to 1 to a ratio of 5.36 to 1. This was done by a series of cylinder-heads that had been cut down by different amounts, so as to reduce the clearance volume by corresponding stages. The device employed for measuring the relative intensities of different detonations and called the bouncing-pin apparatus is shown diagrammatically in Fig. 1.

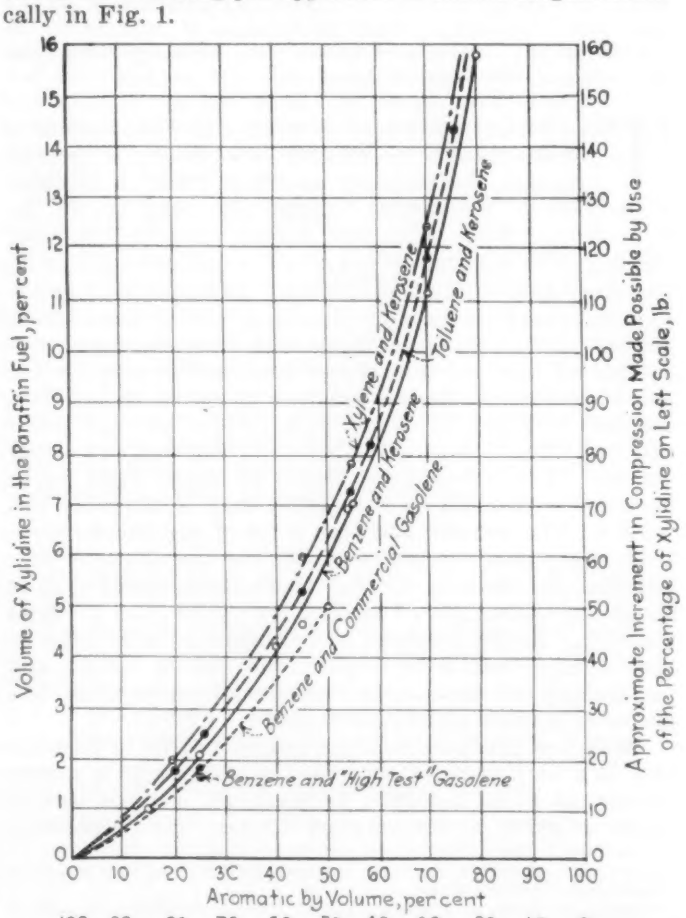


FIG. 2—GRAPHICAL ARRANGEMENT OF THE DATA OBTAINED IN DETERMINING THE DETONATION CHARACTERISTICS OF BLENDS OF AROMATIC AND PARAFFIN HYDROCARBONS

The method used in making the determinations can best be explained by giving a specific example, for which the comparison of a blend containing 45 per cent of benzene and 55 per cent of kerosene with fuels composed of small percentages of xylidine in kerosene was employed. A compression-ratio of 3.87 to 1 was used, so that some detonation would occur, but which was not so violent as to cut down the power of the engine seriously or to cause it to operate in an erratic manner. The fuel under examination was put into one side of the fuel system and the mixing-valve on the engine was adjusted so as to give a maximum of detonation. This adjustment produces close to the leanest possible mixture for maximum power. By trial it was found that 5 per cent of xylidine in kerosene had a slightly less detonating tendency than the benzene-kerosene blend under examination. This fuel was then placed in the other side of the fuel system and its level was adjusted so as to give the point of maximum detonation. The setting of the mixing-valve was left undisturbed throughout the determination so that the compression pressure of the engine would be unchanged. A number of alternate 1-min. runs were then made, with the 5-per cent-xylidine-in-kerosene and the benzene-kerosene blend. The amount of gas evolved in the electrolytic cell during each period was recorded. The output of the generator in volts and amperes was also kept as a matter of record. After three to six runs had been made with each fuel, the benzene-kerosene blend was replaced with 4 per cent of xylidine in kerosene and a second series of runs was made in the same manner. The amounts of gas evolved during the 1-min. runs were then averaged, and the values thus obtained for the xylidine-kerosene fuels were plotted on a coordinate chart having as its vertical axis the amount of gas evolved per minute and as its horizontal axis the percentage of xylidine in kerosene. These two points were next joined by a straight line. From the point at which this line crossed the horizontal line corresponding to the volume of gas evolved by the benzene-kerosene fuel under examination a vertical projection was made to the horizontal scale at the bottom of the chart giving the percentage of xylidine in kerosene. The intersection of this projected line with the bottom scale then gave directly the percentage of xylidine in kerosene that was equivalent in its effect for the supression of detonation to 45 per cent of benzene in kerosene.

Results

The data obtained in the tests on which this paper is based are given in Tables 2 and 3. The averages of the results given in these tables have been used in plotting the curves on the chart reproduced in Fig. 3. Attention is called to the consistency of the data, and to the agreement between the results obtained with like concentrations of given materials. The close checks that were made in different determinations of the detonation characteristics of a given blend indicate that the values as obtained have a high degree of accuracy.

Fig. 2 gives a graphical presentation of the results obtained in measuring the detonation characteristics of blends of aromatic and paraffin hydrocarbons. These data are tabulated and discussed in the previous paper referred to above. From Fig. 2 the rapidly increasing slope of the curves as the percentage of the aromatic constituent is raised may be noted. Thus the curves show that the presence of only a small percentage of an

TABLE 1—PHYSICAL DATA ON THE FUELS USED IN THE TESTS

Hydrocarbon	Kerosene	Commercial Gasoline	"High-Test" Gasoline	Benzene, 90-Deg. Benzol	Toluene	Xylene
Specific Gravity at 15 Deg. Cent. (59 Deg. Fahr.)	0.816	0.734	0.704	0.878	0.860[a]	0.860[a]
Absorption in Cold Sulphuric Acid, per cent	7	5	3	....	....	....
Distillation Temperatures						
First Drop, deg. cent.	186.0	40.0	44.0	74.0	107.0	135.0
deg. fahr.	366.8	104.0	111.2	165.2	224.6	275.0
10 Per Cent, deg. cent.	201.0	65.0	59.0	77.5	108.0	136.0
deg. fahr.	393.8	149.0	138.2	171.5	226.4	276.8
20 Per Cent, deg. cent.	207.0	83.5	68.5	78.7	108.5	136.2
deg. fahr.	404.6	182.3	155.3	173.7	227.3	277.2
30 Per Cent, deg. cent.	212.0	99.0	76.0	79.2	108.6	136.5
deg. fahr.	413.6	210.2	168.8	174.6	227.5	277.7
40 Per Cent, deg. cent.	217.5	111.5	82.7	79.8	108.7	136.7
deg. fahr.	423.5	232.7	180.9	175.6	227.7	278.1
50 Per Cent, deg. cent.	222.0	125.0	89.3	80.1	108.8	136.9
deg. fahr.	431.6	257.0	192.7	176.2	227.8	278.4
60 Per Cent, deg. cent.	227.5	140.0	96.0	80.5	108.8	137.1
deg. fahr.	441.5	284.0	204.8	176.9	227.8	278.8
70 Per Cent, deg. cent.	233.5	157.5	103.0	81.1	108.8	137.3
deg. fahr.	452.3	315.5	217.4	178.0	227.8	279.1
80 Per Cent, deg. cent.	241.0	177.0	114.0	82.0	108.9	137.5
deg. fahr.	465.8	350.6	237.2	179.6	228.0	279.5
90 Per Cent, deg. cent.	253.5	200.0	128.0	85.0	109.0	137.8
deg. fahr.	488.3	392.0	262.4	185.0	228.2	280.0
95 Per Cent, deg. cent.	268.0	219.0	157.0	92.5	109.2	138.1
deg. fahr.	514.4	426.2	314.6	198.5	228.6	280.6
Dry, deg. cent.	291.0	226.0	178.0	....	....	....
deg. fahr.	555.8	438.8	352.4	....	....	....

[a]Approximate.

TABLE 2—DATA OBTAINED IN DETERMINING THE DETONATION CHARACTERISTICS OF VARIOUS BLENDS OF ALCOHOL AND KEROSENE

Determination Number	Compression-Ratio	Spark, Degrees Before Top Dead-Center	Alcohol-Kerosene Blend: Alcohol by Volume, per cent	Kerosene by Volume, per cent	Determined Equivalent Xylidine in Kerosene by Volume, per cent: Individual	Average
74	3.47 to 1	43	15	85	2.25	
75	3.47 to 1	43	15	85	2.30	
76	3.47 to 1	43	15	85	2.40	2.30
68	3.87 to 1	32	25	75	4.60	
69	3.87 to 1	32	25	75	4.60	
70	3.87 to 1	32	25	75	4.60	4.60
53	4.59 to 1	32	35	65	7.40	
54	4.59 to 1	32	35	65	7.15	
55	4.59 to 1	32	35	65	7.00	
56	4.59 to 1	32	35	65	7.10	7.20
50	5.36 to 1	25	50	50	12.40	
51	5.36 to 1	25	50	50	12.70	
52	5.36 to 1	25	50	50	12.80	12.60

TABLE 3—DATA OBTAINED IN DETERMINING THE DETONATION CHARACTERISTICS OF VARIOUS BLENDS OF AN EQUI-MOLECULAR MIXTURE OF ALCOHOL AND BENZENE WITH KEROSENE

Determination Number	Compression-Ratio	Spark, Degrees Before Top Dead-Center	Fuel Blend: Equi-molecular Alcohol and Benzene, by Volume, per cent	Fuel Blend: Kerosene, by Volume, per cent	Determined Equivalent Xylidine in Kerosene, by Volume, per cent, Individual	Average
77	3.47 to 1	43	20	80	2.35	
78	3.47 to 1	43	20	80	2.25	
79	3.47 to 1	43	20	80	2.15	2.25
71	3.87 to 1	32	35	65	4.65	
72	3.87 to 1	32	35	65	4.75	
73	3.87 to 1	32	35	65	4.75	4.75
80	4.59 to 1	32	50	50	8.60	
81	4.59 to 1	32	50	50	8.20	
82	4.59 to 1	32	50	50	8.60	8.50
84	5.36 to 1	25	65	35	13.15	
86	5.36 to 1	25	65	35	13.55	
87	5.36 to 1	25	65	35	13.45	
88	5.36 to 1	25	65	35	13.30	13.40

aromatic hydrocarbon in a paraffin fuel has but a slight effect toward supressing detonation. This is in agreement with the practical observation made by those who have used benzol-gasoline blends that the addition of less than 20 per cent of benzol to a commercial gasoline or a naphtha exerts only a small influence toward causing the engine to give smoother operation. But when benzol is blended with paraffin fuels in larger percentages its effect increases rapidly as its concentration relative to the paraffin fuel is raised. This is due, in part at least, to the greater percentage of reduction in the amount of the paraffin constituent present as the aromatic content of the blend is increased. It will also be observed from Fig. 2 that toluene on the basis of volume is more effective than benzene for eliminating detonation conditions, and that xylene is, in turn, still more effective than toluene for this purpose.

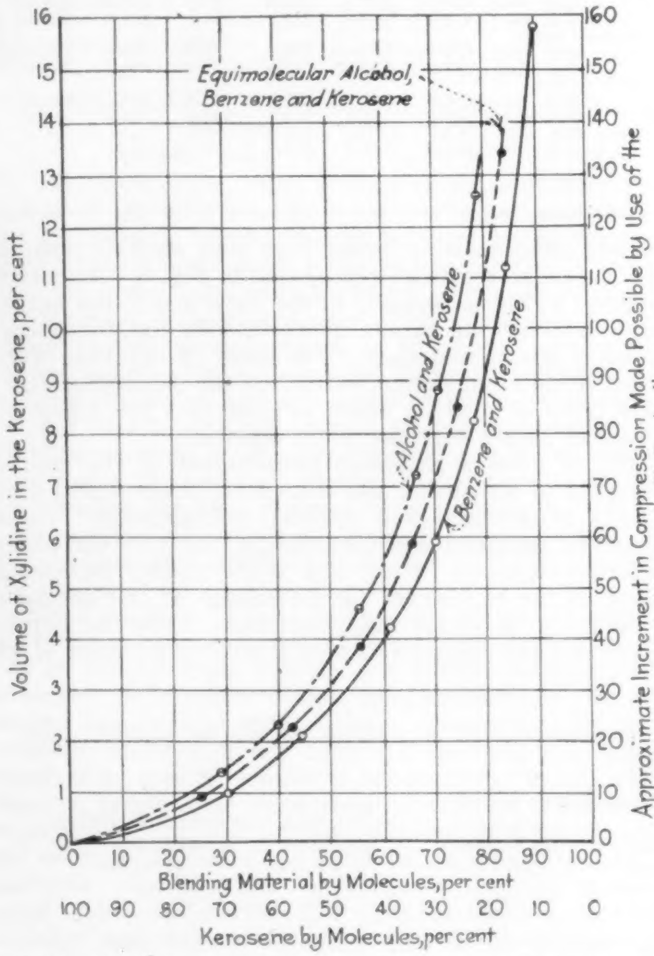


FIG. 3—CHART SHOWING THE EFFECTS ON THE DETONATION CHARACTERISTICS OF KEROSENE OF BLENDING WITH IT VARIOUS PERCENTAGES OF ALCOHOL AND BENZENE

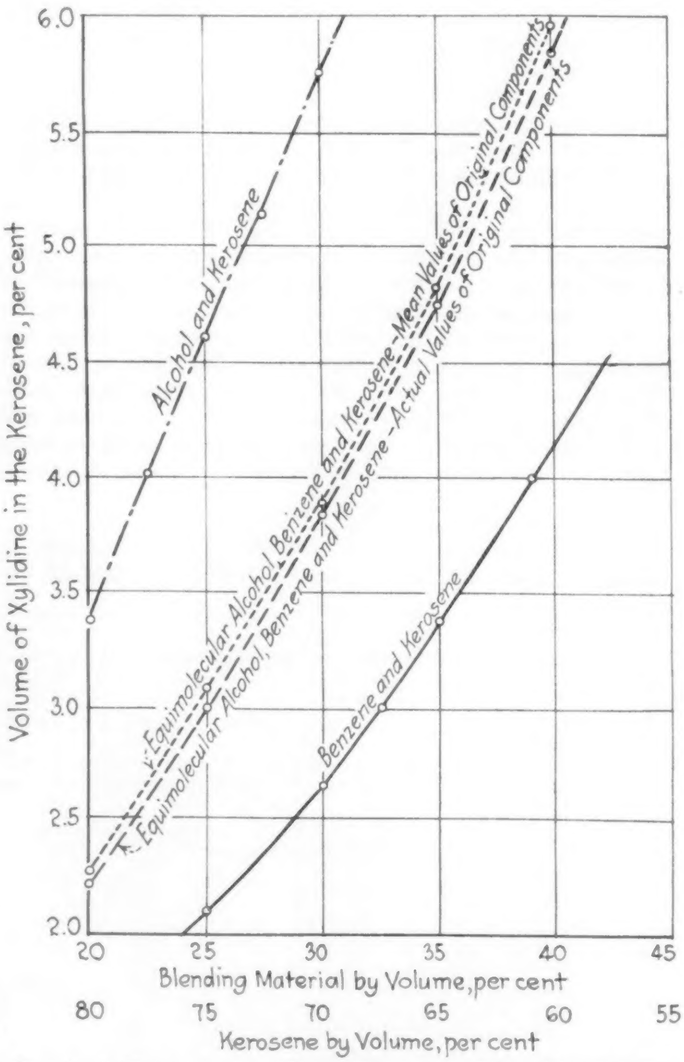


FIG. 4—A BLEND OF TWO FUELS SOMETIMES HAS A GREATER TENDENCY TO DETONATE THAN IS INDICATED BY A MEAN OF THE VALUES OF ITS COMPONENTS

The vertical scale at the right of Fig. 2 shows approximately the increments in compression pressure of the engine that are made possible by the addition to a paraffin fuel of the corresponding percentages of xylidine given on the vertical scales to the left. From the two scales on the charts it will be observed that the addition of 1 per cent of xylidine to a fuel that gives incipient detonation in a certain engine makes it possible to raise the compression of the engine about 10 lb., without any greater detonation being obtained than with the untreated fuel at the original and lower compression. The increment in compression made possible by each per cent of xylidine added to the fuel can only be approximated, but the values given are based upon a number of observations made under practical operating conditions, on engines ranging from the single-cylinder Delco-Light to the 12-cylinder Liberty, over a compression range of from 50 to 160 lb. By referring the curves given on the charts to the scales at the right, an approximation may be obtained of the relative composition necessary to give smooth operation at a corresponding increase above the normal limiting or critical compression of the paraffin fuel alone.

TABLE 4—A COMPILATION OF DATA BASED ON FIG. 3 AND ILLUSTRATING THE OBSERVATION THAT A BLEND OF TWO FUELS SOMETIMES HAS A GREATER TENDENCY TO DETONATE THAN IS INDICATED BY THE CHARACTERISTICS OF ITS COMPONENTS

(1) Blending Material in Fuel on the Basis of Volume, per cent	20	30	40	50
(2) Excess Effect of Alcohol over Benzene in Equivalent Percentage of Xylidine	1.900	3.050	4.650	6.650
Excess Effect of Alcohol-Benzene Mixture over Benzene in Equivalent Percentage of Xylidine				
(3) Actual Value	0.700	1.150	1.700	2.550
(4) Mean Value of Components	0.750	1.200	1.820	2.600
(5) Ratio of Value in Item 2 to That in Item 3	0.369	0.377	0.366	0.384

The data in Tables 2 and 3, together with the benzene-kerosene curve of Fig. 2, are arranged graphically in Fig. 3. This chart shows to a good advantage the relation between the effectiveness of alcohol and that of benzol for the suppression of detonation when blended with a paraffin fuel. It will be observed that on the volume basis alcohol is considerably more effective than benzol for this purpose. Thus, from the chart, 35 per cent of alcohol blended with kerosene produces an effect in suppressing the detonating tendency of the fuel equal to that given by 55 per cent of benzol blended with the same material.

The middle curve of Fig. 3 is plotted from data obtained in determining the effect of blending an equi-molecular mixture of alcohol and benzene with kerosene in the percentages by volume as indicated in Table 3. This equi-molecular mixture was composed of 39.2 per cent of alcohol and 60.8 per cent of benzene by volume. A given volume of the alcohol-benzene mixture contained, of course, equal amounts of alcohol and benzene on the molecular basis. Thus, 100 cc. of the mixture contained 0.675 gram molecules of each of the ingredients, alcohol and benzene. The curve of the detonation characteristics of the fuel obtained by blending this mixture with kerosene should lie somewhere between the curves obtained in a similar way when using alcohol-kerosene and benzene-kerosene blends, respectively. Since the alcohol-benzene mixture contained 39.2 per cent of alcohol and 60.8 per cent of benzene by volume, it is natural to suppose that for a given concentration of this mixture in kerosene the point representing the detonation value of the blend should lie above the benzene-kerosene curve a distance equal to 0.392 part of the differential between similar points on the benzene-kerosene and the alcohol-kerosene curves. Because they are of such small magnitude no account has been taken here of the changes in volume that occur when some of these materials are blended. In making the mixtures used in the tests, each ingredient was measured separately; that is, before being blended. But, while the actual points lie very close to this mean value, it is significant that in every case they are below it. This statement is illustrated by the curves of Fig. 4 and by the data presented in Table 4. The values given in the first three items of Table 4 were taken directly from the curves of Fig. 3, and those in Item 4 were obtained by multiplying the corresponding values in Item 2 by 0.392. The curves in Fig. 4 are based on those in Fig. 3 and on the figures in Table 4.

TABLE 5—RELATIONS BETWEEN AMOUNTS OF XYLIDINE AND ALCOHOL REQUIRED TO IMPART TO KEROSENE LIKE COMBUSTION CHARACTERISTICS FROM THE STANDPOINT OF DETONATION

Xylidine in Kerosene, by Volume, per cent	In Percentages, by Volume, Alcohol	Kerosene	In Equivalent Percentages, by Molecules, Alcohol	Kerosene
1.4	10.0	90.0	29.0	71.0
2.3	15.0	85.0	40.0	60.0
4.6	25.0	75.0	55.7	44.3
7.2	35.0	65.0	67.0	37.0
8.8	40.0	60.0	71.5	28.5
12.6	50.0	50.0	79.0	21.0

It appears, then, that the mixture obtained by blending two fuels of definite detonation characteristics sometimes has a greater detonating tendency than is indicated by the arithmetical mean between the components on the basis of the percentage in which each is present. Attention has previously been called to the fact that in some cases two fuels of similar detonation characteristics, upon being blended, give a fuel that has a very much greater tendency to detonate than either of the ingredients.[6] The results obtained in the tests reported in this paper appear to indicate that this characteristic is common to blended fuels; that is, the detonating tendency of a fuel composed of two ingredients is greater than the average of the values representing the detonating tendencies of the two components taken separately. But this is a point that has not yet been determined accurately for a wide range of different materials.

TABLE 6—RELATIONS BETWEEN AMOUNTS OF XYLIDINE AND EQUI-MOLECULAR MIXTURES OF ALCOHOL AND BENZENE REQUIRED TO IMPART TO KEROSENE LIKE COMBUSTION CHARACTERISTICS FROM THE STANDPOINT OF DETONATION

Xylidine in Kerosene, by Volume, per cent	In Percentages, by Volume, Alcohol-Benzene	Kerosene	In Equivalent Percentages, by Molecules, Alcohol-Benzene	Kerosene
0.95	10.0	90.0	24.7	75.3
2.25	20.0	80.0	42.5	57.5
3.85	30.0	70.0	56.0	44.0
5.85	40.0	60.0	66.3	33.7
8.50	50.0	50.0	74.7	25.3
13.40	65.0	35.0	84.5	15.5

[6] See THE JOURNAL, January, 1922, p. 10.

In Tables 5 and 6 and in Fig. 5 are shown the results obtained by converting the percentages of the fuel ingredients by volume to the molecular basis. The compositions in percentages by volume as given in Tables 5 and 6 were taken from the curves in Fig. 3. In computing the percentage composition of a blend on the molecular basis from its composition by volume the specific gravity and the molecular weight of each of the ingredients were employed. In view of the somewhat wide distillation-range of the benzol used in the tests, the values for which are given in Table 1, a molecular weight of 79 instead of 78 was taken for the benzene. Since kerosene is not a definite compound, and therefore does not have a definite molecular weight, it was necessary to compute an "average molecular weight" for the material. This was done by the method of Wilson and Barnard.[7] For this purpose the distillation data of the fuel given in Table 1 were arranged in the usual type of curve in which the temperature is plotted on the vertical axis against the percentage distilled on the horizontal axis. From this curve the percentages of the fuel distilling in each 10-deg. interval were obtained, and these values were plotted on a chart in which the scale of the vertical axis was in terms of temperature. The average boiling-point of the fuel was taken as the point at which a perpendicular passed through the center of gravity of the area enclosed under this differential distillation-curve cut the horizontal or temperature axis. As determined in this way the average boiling-point of the kerosene used in these tests was 226 deg. cent. (439 deg. fahr.). The approximate molecular weight of the kerosene was computed so as to bear the same proportionate relation to the hydrocarbons next above and below it in the paraffin series as the average boiling-point of the fuel bore to the normal paraffin hydrocarbons occupying like positions with respect to it, dodecane and tridecane. Obtained in this way, which it is recognized gives only a close approximation, the average molecular weight of the kerosene was 178.5.

[7] See THE JOURNAL, November, 1921, p. 313.

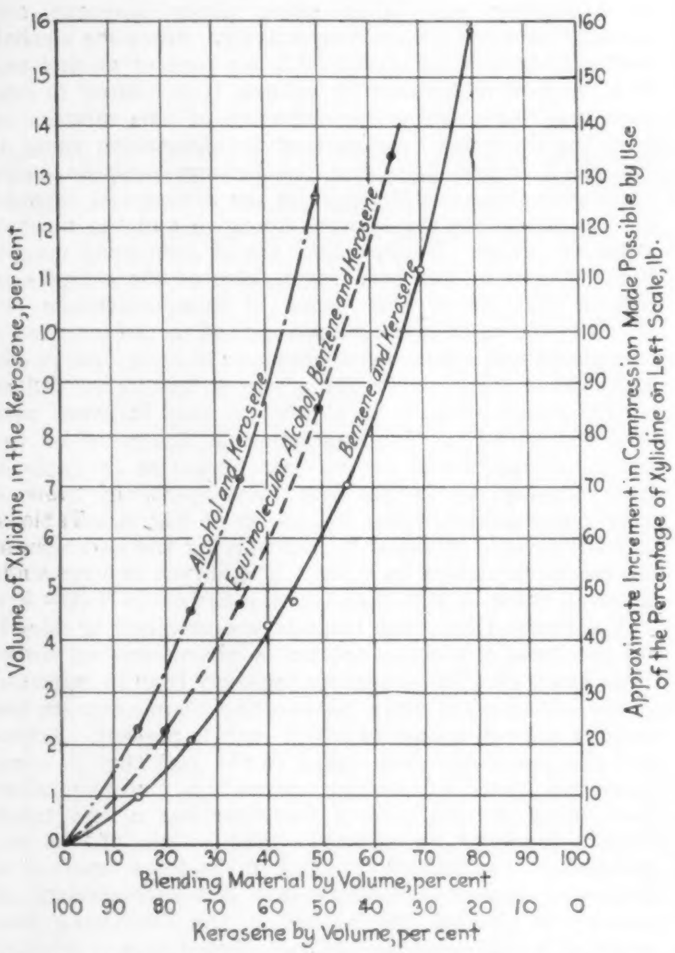


FIG. 5—CHART SHOWING THE EFFECTS ON THE DETONATION CHARACTERISTICS OF KEROSENE OF BLENDING WITH IT VARIOUS MOLECULAR PERCENTAGES OF ALCOHOL AND BENZENE

On the molecular basis there is not such a marked difference between the effectiveness of alcohol and that of benzol for suppressing detonation as shown in Fig. 5 as there is on the volume basis as indicated in Fig. 3. The closer agreement between the effects of the two materials on the basis of molecular concentration is due to the smaller size of the alcohol molecule as compared with that of the benzene molecule. But even on this basis alcohol is still more effective than benzene for suppressing detonation. Thirty-five per cent of alcohol, which, as is indicated above, is equivalent in effect to 55 per cent of benzene on the volume basis, is equivalent to 42 per cent of benzene on the molecular basis.

Since the middle curve in Fig. 5 is based on the results obtained by blending an equi-molecular mixture of alcohol and benzene with kerosene, it is natural to suppose that any point on it should lie half-way between the points occupying like positions on the two outside curves, which were obtained by blending alcohol and benzene separately with kerosene. However, the points on the middle curve do not occupy this middle position; but, as is the case on the volume basis, as shown in Table 4 and Fig. 4, they are uniformly lower than the mean values of the original components, thus showing that in this case a blend of the two ingredients is not so effective for the suppression of detonation as the mean average of the effects of the ingredients would indicate.

THE FARMER'S REWARD

IT is not at all certain that the farmers are rewarded less than the rest of us. In the height of the war period he seems to have done better than the mass of the rest of us; it has been shown that in 1918 the average reward per farmer for labor, risk and management, measured by the prices of 1913, was $826; of factory employes, $725, and of Government employes, $566.

Before the war the farmer was low man in the scale, and he may have returned there, but it is not clear whether he has been properly debited with the value of what he consumes from the product of his farm, including fuel. Taking that into account and also the entirely legitimate differences in the character of clothing and other necessities required, respectively, by urban and rural dwellers, as well as in dwelling site values, ground rent of homes, the farming group may not seem so badly treated as some people might think after all.

This does not militate in the least against the desirability of absolutely fair and just treatment of the farmer, together with such governmental consideration as may not be special privilege rather than primarily for the common gain of all concerned.—Robert Luce in *Economic World.*

A Method of Developing Aircraft Engines

By Capt. George E. A. Hallett, U. S. A.[1]

Semi-Annual Meeting Paper *Illustrated with* Photographs

THE general method of procedure taken by the Air Service before beginning the actual design and construction of the necessary types of aircraft engine is outlined and the four steps of the development subsequent to a very complete study of existing domestic and foreign engines are stated.

After checking over the layouts, if all the details are agreed upon by both the designer and the Engineering Division, the contract is placed, usually for two experimental engines, and the construction work is begun.

Acceptance tests are made to demonstrate that the engine is capable of running at normal speed and firing on all cylinders. These are followed by the standard performance test made on the dynamometer at McCook Field. The results of the latter test determine whether the engine can enter the 50-hr. endurance test. The engine is then torn-down and inspected for wear. Suggested modifications are embodied in reconstructed engines which eventually fulfill the requirements. Descriptions of the various tests are given and commented upon.

THE development of an aircraft engine is defined as the work which is done between the time of making the initial scale layout and the time when the engine is ready for production. The purpose of this paper is to describe briefly the method pursued by the Engineering Division at McCook Field in accomplishing this work. Before entering into a description of the methods of aircraft engine development the need for the many types will be explained briefly and what are the general steps to be taken before beginning the actual design and construction of these types.

At the close of the war, very few fully developed types of engine were available to our Air Service. We had a bombing engine which was good for various two-seater airplanes, but we did not have entirely satisfactory pursuit, training or heavy-bombing engines. The operations group of the Air Service promptly prepared a set of performance specifications for 16 types of aircraft for as many military uses. Working from these specifications, a similar list was prepared of the sizes and types of engine that would be needed in this program, with the idea of making them available as rapidly as possible; and a very complete study of existing domestic and foreign engines was begun. First, standard test methods were laid down. Next, standard test-report outlines and contents were decided upon. Then domestic and foreign engines were tested in two ways, (*a*) by a so-called standard test that included thorough and accurate measurements of all phases of the engine's performance and in some cases, (*b*) by a 50-hr. test for the purpose of comparing the durability of the engine with that of others. As this work got well under way, we started to work out improvements for existing domestic types and arranged with suitable contractors for the design and construction of new types that appeared to be needed. In a very few cases where no contractors could be interested the designs were laid down at McCook Field.

The study of existing domestic and foreign engines provided valuable data on such characteristics as brake mean effective pressure, horsepower per cubic inch piston displacement, pounds of engine per cubic inch of displacement, comparative freedom from vibration of different types and comparative durability. It was noted that the fuel-consumption was controlled largely by the compression-ratio, the type of carbureter and the type of manifold, provided the engine showed reasonable mechanical efficiency and brake mean effective pressure. These data on the characteristics of the best existing engines served as a fairly good guide in regard to what types of design and construction would prove most satisfactory for incorporation in the engines to be developed, and it should be noted particularly that since all these measurements were taken on the same standard equipment and by the same standard methods, the results were unusually comparable.

Development Procedure

In general, the steps in the development of engine types have been as follows:

(1) The design and layout are started in most cases by civilian contractors

(2) The Engineering Division, upon receipt of these layouts, generally checks the stress and weight calculations

(3) The engine is checked from a standardization and installation point of view; in other words, the gasoline, oil, water, throttle and electrical connections must be standard and located at points that will be accessible after the engine has been installed; provision must be made for mounting the standard electric starter, gun synchronizers, gasoline pump, tachometer drive and electric generator. Much serious trouble in airplane design and engine maintenance is thus avoided

(4) The layout is checked to see that it has embodied the latest experience of the Engineering Division and its contractors in design and constructional details

The Engineering Division acts as a clearing-house for approved practice in design and construction. For example, the good points of an engine designed by the Engineering Division or a civilian company can be applied to an engine designed by some other company. This practice has been of great benefit when engines have been designed by the various civilian contractors as well as when designed by the Engineering Division.

After checking the layouts and when all the details have been settled to the satisfaction of the designer and the Engineering Division, the experimental contract is placed and the construction work started. The contract

[1] S.M.S.A.E.—Chief of powerplant section, engineering division, Air Service, Dayton, Ohio.

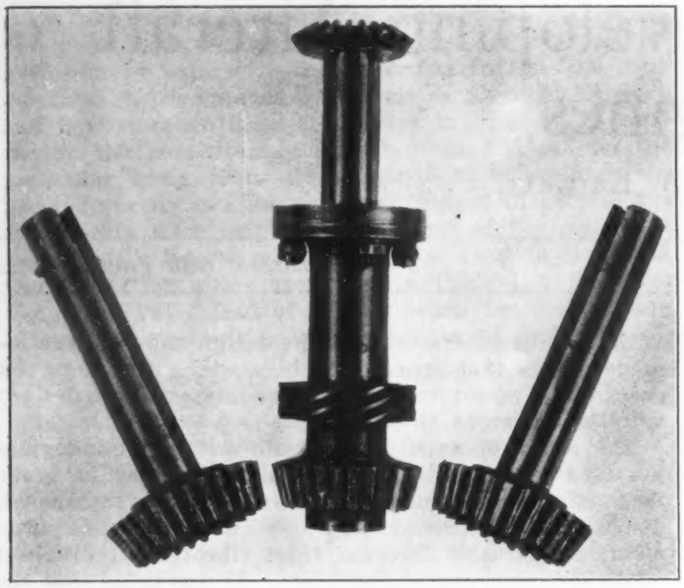


FIG. 1—TYPICAL SET OF CRANKSHAFT GEARS THAT HAVE BEEN DAMAGED BY WHIPPING OF THE CRANKSHAFT

generally calls for at least two engines, inasmuch as it has been found that much delay is avoided by having an extra engine ready for test in case of an accident occurring to the first one; however, when the design is considered unusually experimental, the contract generally is made for one engine with a rather large allowance for spare parts, which amounts almost to a second engine unassembled. After the construction of the engine is started, the detailed inspection generally is carried on by the contractor, but usually the plant is visited and the work looked over during the most interesting stages of construction.

ACCEPTANCE TESTS

After the engine has been completed, the acceptance test is run, sometimes at McCook Field and sometimes at the contractor's plant, according to where the work can be done most conveniently. The nature of these tests is generally little more than a demonstration that the engine is capable of running at normal speed and firing on all cylinders. The more experimental or nonconventional the engine is, the less severe are the tests. This is due to an attitude on our part of being responsible for the design, since it has been checked so carefully

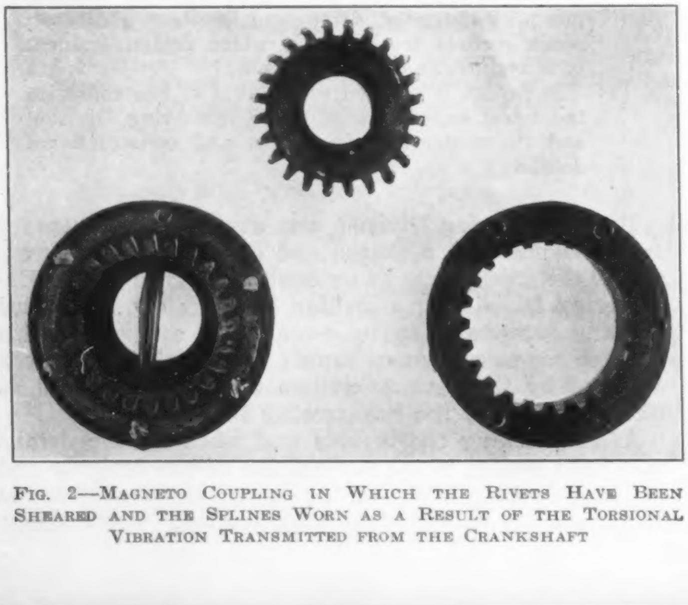


FIG. 2—MAGNETO COUPLING IN WHICH THE RIVETS HAVE BEEN SHEARED AND THE SPLINES WORN AS A RESULT OF THE TORSIONAL VIBRATION TRANSMITTED FROM THE CRANKSHAFT

from the start and since many basic changes are often incorporated. The acceptance test generally shows whether the engine has been constructed and assembled in a satisfactory manner and the engine is accepted or rejected on the basis of the results of this test. It undergoes the standard performance test at McCook Field.

If the cylinder construction of the new engine is not of a well-proved type, the construction of the complete engine usually is held up pending the construction of one cylinder, the making of thorough tests of it on one of the universal test engines. These single-cylinder tests often show many troubles but, fortunately, most of them can be remedied by modifying the design or methods of construction. Frequently changes in the design of the valves, valve-guides or valve-gear are found necessary. Sometimes the compression-ratio proves too high or unnecessarily low, and the best location and number of spark-plugs as well as the proper spark-timing and piston clearance have to be ascertained. If the results with the single cylinder should prove too bad, the entire project would be held up indefinitely pending the design and development of a more satisfactory cylinder; or perhaps abandoned. Generally the results can be made satisfactory and the work proceeds.

We find that the single-cylinder tests are very satisfactory in everything but fuel-consumption. Fuel-consumption readings are only comparable with other single-cylinder results and are not comparable with results to be obtained from multi-cylinder engines. The writer is responsible for the use of these single-cylinder engines for this work and is endeavoring to carry the idea of testing "by units" much farther. In newly designed engines, the fuel-pumps, oil-pumps, water-pumps, ignition systems and carbureters can all be thoroughly tested in an accessories laboratory entirely independently of the rest of the engine. In one of the engines designed at McCook Field an entire gearcase assembly was tested to determine whether the complicated lubricating system was satisfactory. This was done by mounting it in the accessories laboratory and driving it at normal speed by an electric motor, using a thin oil of a viscosity similar to that of the Air-Service specification oil when at the working temperature. Faults in the oiling system were found much more quickly and cheaply in this way than by testing the assembly on a complete engine. Obviously, if enough parts of the engine can be tested separately at small expense and risk, the complete engine is far more certain of satisfactory results when assembled, and many costly wrecks can be avoided. It is believed that much more progress can be made along this line.

THE STANDARD TEST

When the completed engine is placed on the dynamometer at McCook Field ready for its standard performance tests, assuming that it has not been placed in good running condition previously, the procedure is about as follows: The engine is motored over with plenty of good oil and water circulating through it until the horsepower needed to motor it over at normal speed bears a predetermined relation to the horsepower to be expected from the engine, thus assuring satisfactory and standard conditions within the engine. Carbureter chokes are selected that will produce a satisfactory manifold depression when the engine is being motored over at normal speed. We do not deceive ourselves by running engines with large chokes and low manifold-depressions that are impractical in flight; therefore, the manifold vacuum is held between 1 and 2 in. of mercury. After this, the jets can be adjusted so that they meter an amount of fuel

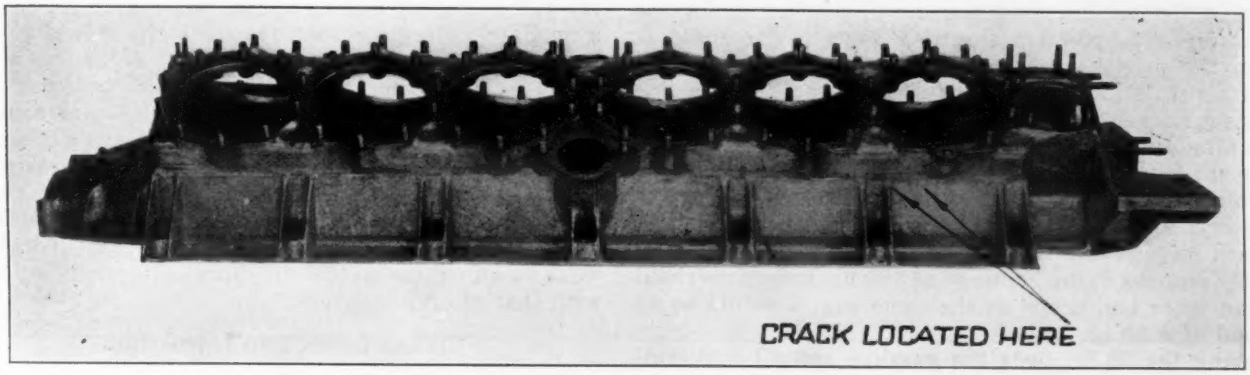


Fig. 3—Crankcase That Cracked in Several Places While Undergoing Tests

in accordance with that to be expected per horsepower-hour, thus insuring a carbureter setting and mixture-ratio easily within the running range. Further adjustments must be made by actually running the engine under its own power, but the preliminary steps avoid much guessing at the start. Many other precautions are taken at this stage of the work but they are covered in detail in Engineering Division reports.[2]

As soon as the engine is running satisfactorily, steps are taken to improve its performance, if possible, and it is then put through the standard test planned to provide all the desired data on the performance of the engine. These include the obtaining of full-throttle-power and propeller-load-power curves, which give a good indication of the fuel-consumption in throttle flight; measuring the fuel and oil consumption; and the data needed by the airplane designer such as rate of water flow, water-temperature rise and the like. In case of failure of the engine during the standard test, parts that are affected are studied carefully and, if practicable, they are replaced and the test is continued. Fortunately, most of the engines we have developed have been able to get through the standard tests and at least get started on the long endurance test that follows. At the close of the standard test the engine is disassembled and inspected with the idea of ascertaining whether it is in shape, or can be put in shape, to start on the endurance test. If at the end of the standard test the engine is not in shape to start on the endurance test, it may be repaired, redesigned or abandoned, according to the results shown in the standard test.

[2] See Engineering Division reports Nos. 1506 and 1507.

The 50-hr. test is a standard method of comparing the durability of engines. It is a first step in developing an engine toward satisfactory durability. These tests are nearly always made on a torque-stand with a propeller

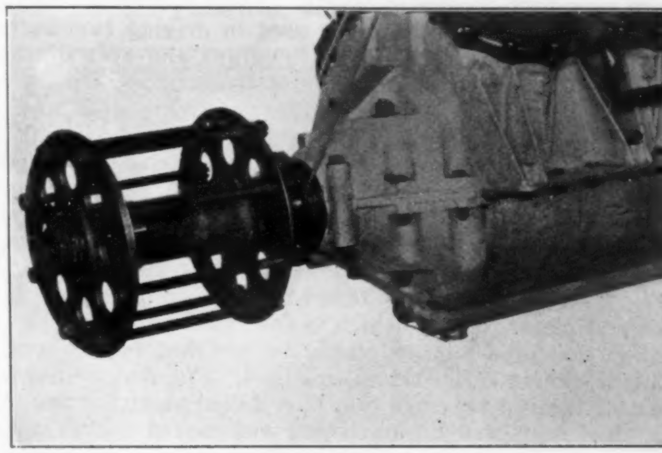


Fig. 4—Portion of a Crankcase in Which the Design of the Ribs in the Upper Half Was Modified To Provide Additional Stiffness

providing the load, and are based on the normal speed and power of the engine. A propeller is selected that will permit the engine to run at its normal speed with the throttle wide-open and absorb the normal rated power of the engine under these conditions. The 50-hr. test consists of 10 non-stop runs of 5-hr. duration, each run beginning with ½ hr. at full throttle and full speed with

Fig. 5—Faulty Distribution of the Load Caused the Failure of These Connecting-Rod Bearings

the propeller as described above and continuing for the remaining $4\frac{1}{2}$ hr. when throttled down to the speed at which the propeller absorbs 9/10 of the power at full throttle. Due to the fact that in an airplane the engine generally is run for only 5 or 10 min. at anything approaching full throttle and is then throttled to a much lower speed when the airplane is taken up to an altitude at which the power output is greatly reduced because of the decreased density of the atmosphere, the 50-hr. test is much more severe than 50 hr. of flying; in fact, many Liberty engines flying as many as 180 hr. before overhaul are not in as bad shape as the same engine would be at the end of a 50-hr. test.

During the 50-hr. tests the gasoline and oil consumption and power fluctuations are noted carefully and the valves are watched for heating; in fact, the entire engine is watched carefully and an accurate record is kept of all troubles and of work that becomes necessary. Many troubles are encountered due to torsional or other periodic vibrations in the crankshafts, especially in those which are over four throws in length. These vibrations result in the breakage of the gears used in driving camshafts or magnetos, of the magneto couplings and even of the magnetos themselves. Sometimes these troubles require several modifications before they are eliminated completely.

Fig. 1 shows a typical set of camshaft drive-gears that have suffered from such "crank-whip." A Hispano-type magneto-coupling in which the rivets have been sheared and the splines worn, due solely to torsional vibration transmitted from the crankshaft, is illustrated in Fig. 2. Fig. 3 shows a crankcase that was found to be cracked in several places, one of which is indicated by the arrows. A portion of a similar crankcase modified to overcome this weakness is illustrated in Fig. 4. Fig. 5 shows some connecting-rod bearings that have failed partially, due to the bad load distribution which was caused by unsatisfactory connecting-rod design. The bearings out of the same engine after another 50-hr. run with connecting-rods of a new design appear in Fig. 6. As will be noticed by comparing Figs. 5 and 6, the bearings in the latter show great improvement. Bearing No. 5 was injured by the breakage of a defective crankshaft. Fig. 7 illustrates some exhaust-valves that have suffered excessive wear on their tips and stems and from burning. In this case all these troubles were traceable to a defective rocker-arm design that imposed too much side-thrust on the valve-stems, an extreme instance of this type being illustrated in Fig. 8. Fig. 9 shows the valve-stem-guide bushings and Fig. 10 is reproduced from a photograph of the valves from the same engine after a 50-hr. run with a modified valve-gear that overcame the difficulty.

Some engines prove hard on spark-plugs, crack their water-jackets under the severe testing conditions, burn their valves and break minor parts such as valve springs, rocker-arms, oil lines and the like. Occasionally we have a serious wreck before the completion of the 50-hr. test; but in every case we endeavor to get 50 hr. of running if we possibly can, since this is not an acceptance test but a method of obtaining 50 hr. of a standard kind of wear on an engine so that its durability can be compared with that of other engines.

Tear-Down and Inspection

After the 50-hr. test, the engine is torn-down and inspected carefully. Most parts are inspected both before and after cleaning. All parts are removed to the photographic room where they can be inspected more accurately and can be photographed conveniently. The party that inspects the torn-down engine generally consists of the writer, a designing engineer, the test engineer assigned to the engine in question, the mechanic who has had the engine in charge and other interested persons. As they are examined the parts are checked off systematically from a printed list and the condition and appearance of each part is passed upon. We find that the appearance of parts means much. An agreement concerning each piece is usually reached before passing to the next one.

The help of the Materials Section at McCook Field has proved invaluable in arriving at the reasons for breakage and other kinds of material troubles. A decision as to what will be done about improving the part is withheld until after the receipt of the Materials Section's report. The results of the inspection or conference on the disassembled engine are noted and an agreement is reached with the designer or builder of the engine as to what should be done to meet the criticisms and recommendations. If the results warrant it, arrangements are made with the constructor to construct a modified engine in accordance with the recommendations made after the last inspection. We are glad to say that the engine, when reconstructed, accepted, given a check-run and the 50-hr. test, generally completes the test in a satisfactory manner. It is then torn-down again and inspected. When the need of additional improvements becomes apparent, they usually are made.

The photographs of parts of the first engine that were taken while the engine was torn-down at the end of the 50-hr. test are exceedingly useful for making compari-

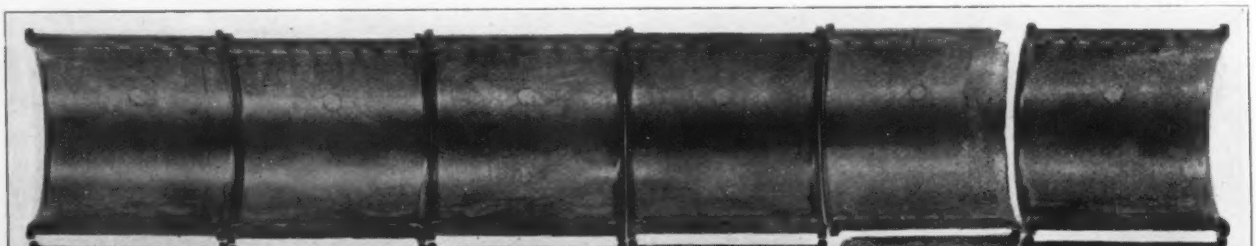


Fig. 6—An Improved Form of Connecting-Rod Bearing

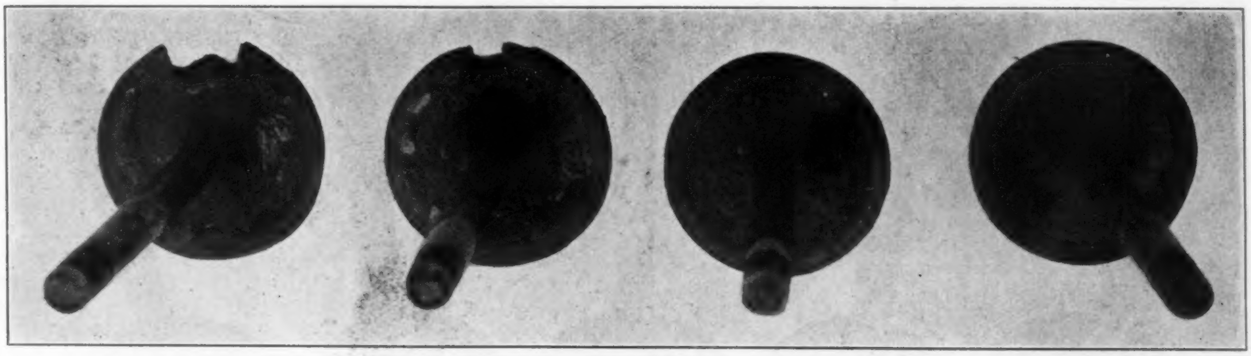


FIG. 7—EXHAUST-VALVES THAT WORE EXCESSIVELY ON THEIR TIPS AS A RESULT OF A DEFECTIVE ROCKER-ARM DESIGN

sons with the observed condition of the second engine, inasmuch as many different engines are handled by our organization and we may not carry in mind perfectly the conditions found in the engines from one test to another. After a sample engine has gone through one of these 50-hr. tests satisfactorily, it generally is considered good enough to warrant the purchase of 8 or 10 more for experimental use.

FLIGHT AND SERVICE TESTS

The first of these engines is put into flight test as promptly as possible, because usually an additional crop of troubles shows up when a new engine gets into the air for the first time. Such troubles as improper venting of the carbureter float-chamber may develop in flight because of changes in the relative pressures in the float-chamber and the throat of the carbureter while the airplane is moving at high speed. The distribution of the mixture may prove bad in sharp maneuvers, even though it may be satisfactory in normal flight. Occasionally, troubles in the water-circulation system appear when the airplane is in abnormal positions. Propeller-hub troubles are encountered occasionally, but the most troublesome point is vibration. Engines that seem to run smoothly on the test-block will shake badly in flight and cause trouble in the surrounding structure. Sometimes the accessibility of an engine does not prove to be as good after it is installed as it seemed to be on paper and on the test-block.

When an engine has reached the point where it has passed the 50-hr. test and is working satisfactorily in the air, and a few others of the same type are in use in the Air Service, it is time to try much longer tests of 100, 200 or even 300 hr. duration. These are of the nature of refining. The remaining weak points which did not show up in the 50-hr. test will begin to appear in the longer test and generally can be corrected then. It usually is possible to develop the engine so that it can run at higher speeds, higher compression-ratios or higher brake mean effective pressures, thereby effectively reducing the

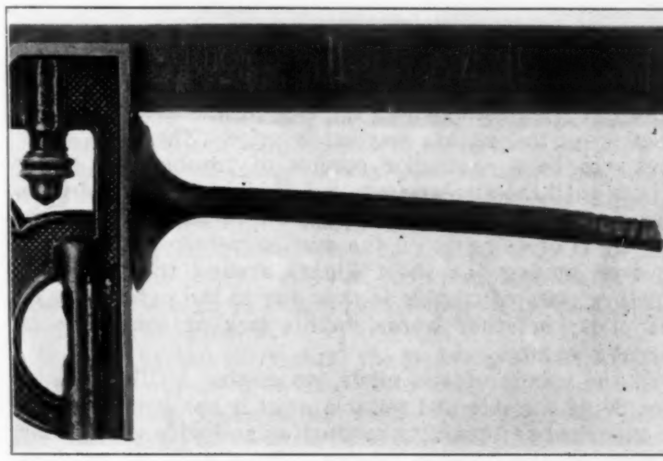


FIG. 8—AN EXTREME INSTANCE OF THE ILL EFFECT OF THE SIDE-THRUST THAT WAS IMPOSED UPON AN EXHAUST-VALVE DUE TO IMPROPER DESIGN OF THE ROCKER-ARM

specific weight of the engine without increasing the difficulties of production or the cost, and at the same time insuring better fuel economy.

When a new type of engine is sent to some field where it has been unknown, many troubles may be encountered which are due to the fact that the personnel is unfamiliar with that particular engine. These might not have shown

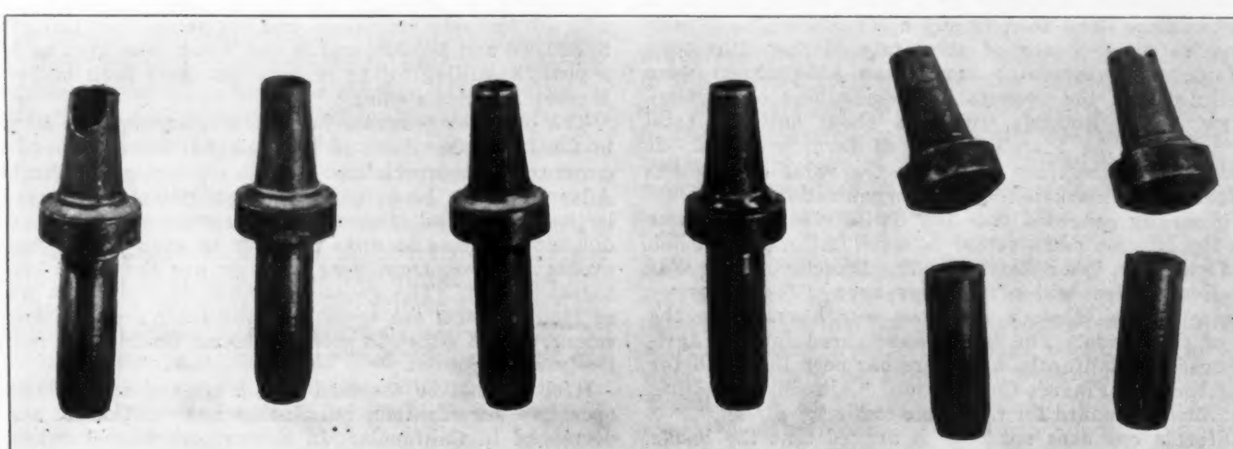


FIG. 9—VALVE-STEM-GUIDE BUSHINGS TAKEN FROM AN ENGINE AFTER IT HAD BEEN TESTED AT MCCOOK FIELD

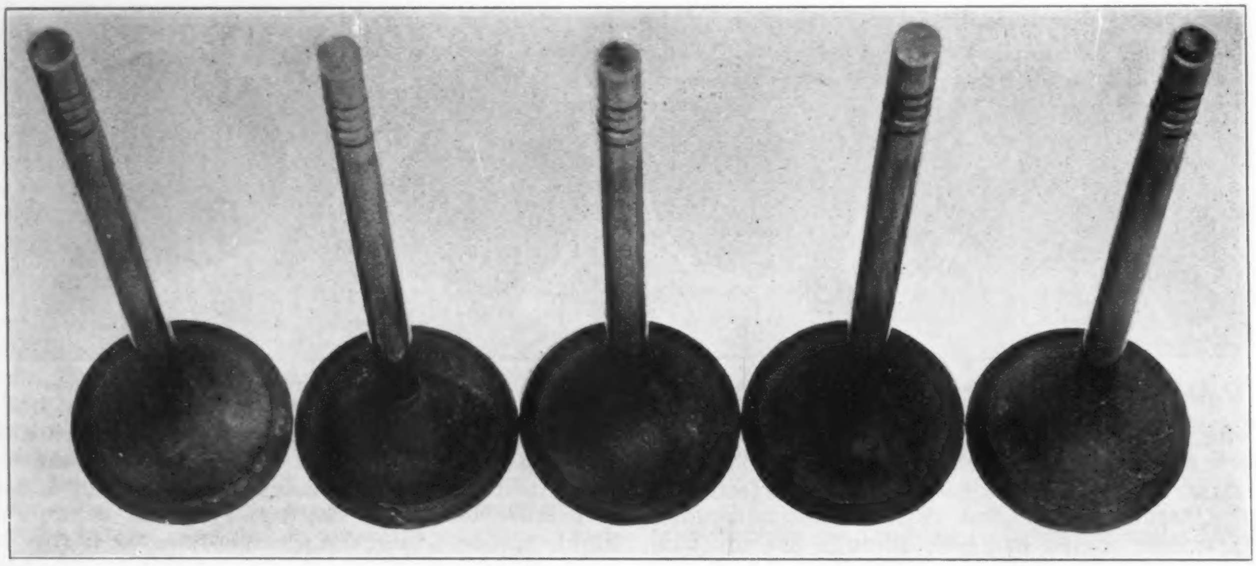


FIG. 10—VALVES OF THE SAME ENGINE, FROM WHICH THE BUSHINGS SHOWN IN FIG. 9 WERE TAKEN, AFTER IT HAD BEEN SUBJECTED TO A 50-HR. TEST WITH A MODIFIED VALVE-GEAR

up during the tests at McCook Field. The maintenance conditions and methods at such a field are likely to be different from those where the engine was developed, and troubles of this nature may easily arise. Then, too, there seems to be a particular species of trouble that never occurs until the airplane gets out of sight of the airdrome. For example, parts will work loose that had never shown any signs of doing so on the dynamometer or block tests or even during the short flights around the airdrome. Another class of trouble is that due to the psychology of the pilot; in other words, to his lack of confidence in the new engine.

In the minds of the pilots, no engine is likely to be considered durable and reliable until it has gone through an experience of quantity production and wide use similar to that of the Liberty engine. Practically all engines have had their troubles when first put into use, and the Liberty engine is no exception. We are informed that due to its more hurried development the first 250 Liberty engines developed cracks in their crankcases; that the first 750 had light crankshafts, which occasionally broke, and weak connecting-rods. Since these weaknesses were rectified the engine has had such wide and satisfactory use that the pilots have long ago forgotten that phase of its development, and it now enjoys their greatest confidence.

In addition to developing complete engines, much work has been done in perfecting accessories that are not really parts of the basic engine but are fairly general in their application, such as carbureters, spark-plugs, piston-rings, valves and pistons. The results of this work have been applied to whatever engines could be benefited and have produced considerable standardization as well as improved performance. Between January 1919 and the present time development has been started on 23 engines. Of these two which were failures have been abandoned. Sixteen have been successes, and five which are not yet complete give promise of being successful.

COOPERATIVE FARM MARKETING

COOPERATIVE farm associations are not new, nor are they confined to California. They exist in various forms in every State in the Union. In California, however, their activities have been developed to a wider extent and on a more substantial basis than in any other State. Last year, according to an estimate of the State Market Director's office, farm products worth more than $250,000,000 were marketed through the cooperative associations of farmers who grew them. Roughly, this was about half the total market value of the State's output of farm products. It exceeded by many millions of dollars the value of products in any other State marketed by farm organizations.

It is generally conceded that the California farmer came through the readjustment period in much better shape than did the farmers in other districts. The Director of the War Finance Corporation said a few days ago: "The times we have been passing through since the war have shown the solidity of California. The banks have cared for the agricultural needs of California, and there has been little call for help upon the War Finance Corporation. California is setting an agricultural standard for the whole country."

In California one does not hear it argued that the banks should be run by "dirt farmers." Membership in a cooperative is limited to actual growers of the crop about which the organization is formed, but for sales managers, and other employes in the financial departments of the business, experienced business-men are hired, and they are not paid the wages of ordinary field-hands. Salaries of managers, who are business specialists and not necessarily farmers, run to $20,000 and $30,000, and in one instance as high as $50,000 a year, according to the reports that have been made to the Market Director's office.

The greatest progress in the farm cooperative movement in California has been in those organizations formed about crops requiring specialized skill in producing and marketing. Adversity has been the strongest stimulant toward collective action, and in general the growers who have the most difficult problems to solve in order to keep their crops on a profitable basis from year to year are the ones who hold together best. The greater number of California growers of food products are specialists and their organizations represent unified effort to solve problems common to comparatively small groups.

It should not be assumed that a cut-and-dried plan of cooperative organization, guaranteed against failure, has been developed in California. In a way, collective marketing is still in the experimental stage.—Monthly Review of San Francisco Mercantile Trust Co.

New System of Spring-Suspension for Automotive Vehicles

By H. M. Crane[1]

Semi-Annual Meeting Paper *Illustrated with* Diagram and Photograph

THE author indicates what the history of spring suspension has been but discusses only the conventional type of four-wheeled design in which the front wheels are used for steering and the rear wheels for driving and braking. The problem of front-axle spring-suspension is mentioned, but that of proper rear-axle spring-suspension, especially for passenger cars, is discussed in detail because it is a much more difficult one.

The advantages of the Hotchkiss drive for shaft-driven cars and some of its distinct disadvantages are stated, shaft-driven, rear-axle mountings being commented upon in explaining the factors that influenced the design of the spring-suspension device developed by the author. The advantageous features of this device are enumerated, inclusive of the effects of tire reactions.

THE first wheeled vehicles of which we have any record were not provided with any special spring members for the cushioning of road shocks. They were not entirely springless, however, for a certain degree of cushioning effect was furnished by the natural resilience of the materials used in the construction of the various parts. There are many horse-drawn wagons still in use today which embody this type of design. As road transportation became more highly developed and speeds increased, vehicles were produced in which the spring action of the various parts was increased by modification of the design, but still without the use of special spring members. The buckboard and the stagecoach are instances of this stage of development.

Near the end of the last century, however, the art of spring suspension, using steel leaf-springs of various forms, had reached a considerable degree of development, at least for horse-drawn vehicles. The early designer of automotive vehicles therefore had a valuable fund of information on this subject with which to work. This information could be used only in conjunction with the working out of an entirely new set of problems, imposed by the fact that the automotive vehicle is propelled, stopped and steered by means of its wheels. For the purposes of this paper, it will be necessary only to discuss the simple and conventional type of four-wheeled design, in which the front wheels are used for steering and the rear wheels are used for driving and braking.

In front-axle design, it is almost universal practice to pivot the wheels for steering by mounting the short stub-axles or knuckles, on which they revolve, on the ends of the axle by substantially vertical hinge-pins. The axle itself requires only to be connected to the frame in such a manner as to remain in a plane approximately at right angles to the longitudinal center-line of the chassis and so that a linkage can be arranged between the steering-gear, which is fixed on the chassis frame, and the steering connection on the axle, in a way that will allow of the axle moving relatively to the frame without tending to alter the direction of the front wheels. These requirements can be met fairly well by using a pair of semi-elliptic springs, fixed at one end and shackled at the other and, due to its simplicity, this is the favorite arrangement today. As I will explain later, this system is not ideal from the point of view of absorbing road shocks by the springs. It depends to a considerable extent for its success upon the pneumatic or other cushion tire, as well as on the fact that, in passenger cars, where easy riding is important, the passengers rarely are carried on the chassis at a point forward of the center of the wheelbase.

Rear-Axle Spring-Suspension

The problem of a proper rear-axle spring-suspension, especially for passenger cars, is a much more difficult one. Like the front axle, the rear axle must be held in its correct position with respect to the frame, but it also must be supported in such a way that the rear wheels can drive or stop the car as required. Furthermore, the connection between the axle and the frame should be designed to minimize the transmission of road shocks from the former to the latter. A brief outline of the history of automotive design in this regard will help to make clear the questions involved.

In most of the early cars, the power was transmitted to the rear wheels by chains, either a single central chain or double side-chains being used. Let us consider only the latter type of construction, which clearly illustrates the principles of design. Rigid distance-rods are used between the axle and the frame. These are required to preserve the correct center distance between the driving and driven sprockets, and also serve to maintain the axle in a proper position with relation to the frame. Semi-elliptic springs, shackled at both ends, support the body load and also act to hold the axle laterally. In driving and braking the vehicle by the wheels, torque reactions are set up. In the type of design under discussion, the reactions due to driving or the use of a transmission brake are taken up in the jack-shaft housing. The reactions from the rear-wheel brakes are taken up in the springs sometimes, but more often by the distance-rods, which are then made stiff enough to act as torque-arms. In the chain-driven car, the rear axle is relatively light and has therefore no great tendency to produce shocks in the vehicle structure. This is not the case with the much heavier axle of the shaft-driven car, and this fact must be recognized in the design of the latter type if the best results are to be obtained.

The early shaft-driven cars tended to follow chain-driven designs, using distance-rods, and adding a torque-arm attached to the rear-axle housing and supported at some point on the frame to take care of the torque reactions arising from driving and braking. The springs were used only as supporting members, being usually mounted so as to turn freely on the axle, and being shackled at both ends. In striving for greater simplicity and decreased weight, it was found that the distance-rods could be dispensed with and their functions performed

[1] M.S.A.E.—Consulting engineer, New York City.

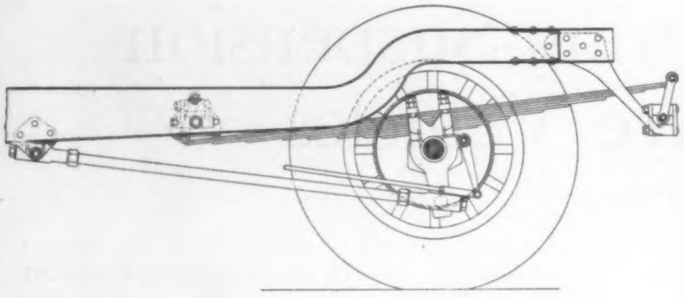


FIG. 1—SCHEMATIC DRAWING OF A RECENTLY DEVELOPED SPRING-SUSPENSION

by the front halves of the rear springs, the shackles at the front ends being abandoned and plain pivot-pins used in their place. A further move in the direction of simplicity led to the Hotchkiss drive, in which system the rear axle is entirely controlled by the rear springs. The springs support the load, maintain the axle in correct relation with the frame and absorb all torque reactions, whether due to driving or braking.

The Hotchkiss drive for shaft-driven cars has some very manifest advantages. It is light in weight and has a minimum number of parts to wear and rattle. It gives a cushioned resistance to suddenly applied driving and braking loads. Moreover, if the springs are properly proportioned, it maintains the center-line of the pinion shaft substantially parallel throughout the range of axle movement. This latter feature is of an importance not often recognized. If a rear-axle torque-arm is used, whenever the axle approaches or recedes from the frame the axle housing must rotate through an angle depending upon the amount of movement and the length of the torque-arm, the angle being greater the shorter the torque-arm is. If there is no lost motion in the parts, the pinion shaft must rotate to an amount depending on the angle through which the rear-axle housing turns, and on the gear-ratio. This would not be a serious matter if the pinion shaft alone were involved but, when the car is being driven by the engine, the pinion shaft is connected through the driving-shafts to the engine flywheel, which is a part of the system designed to turn as nearly as possible at a constant angular velocity. This combination of circumstances is capable of producing shocks in the driving members which are far greater than will be believed readily by those who have not observed this action.

The Hotchkiss drive has also some distinct disadvantages. When the rear springs are sufficiently flexible to give good riding, the resistance to torque reaction is too soft, with the result that, especially in lighter cars, the rear axle tends to jump when under heavy driving or braking loads and on soft or rough roads. From the point of view of good riding it leaves much to be desired. It furnishes a sufficient degree of flexibility in a substantially vertical direction, but is far too rigid otherwise.

Let us consider any of the present common types of shaft-driven rear-axle mounting, either of the Hotchkiss design or of designs using torque-arms or distance-rods, or both in combination. In practically all of these arrangements, the axle is positioned rigidly with respect to the frame in a longitudinal direction. If the weight of the axle were relatively small, this rigidity would not be a matter of serious moment. The fact is, however, that the weight of the rear axle and wheels of most shaft-driven cars approaches 10 per cent of the total loaded weight of the vehicle. The ideal of riding is to have the spring-borne parts of the vehicle, the frame, the power-plant, the body and the load, follow the major contour of the road surface without being affected by minor undulations. It would be desirable to have this condition apply also to the parts below the springs, the wheels and axles, and the pneumatic tires help to attain this result, but there is a limit to their ability in this direction unless made of extreme dimensions. Practically, the axles and wheels follow a very irregular course, with rapid and frequent changes of direction in a vertical plane. To change the direction of motion of a moving mass force must be applied and, if the mass is great and the change in direction sudden, that force must be large. In most cases the forces in question are the reaction of the road surface through the tire and the horizontal attachment between the axle and the frame such as distance-rods or springs. This latter connection is usually extremely rigid, and so it follows that the only cushioning for the shocks occasioned by these forces is furnished by the tires and what little resilience there may be in the road surface. Spring-cushioned distance-rods have been tried with a view to ameliorating this condition, but the cure in this form has seemed to be worse than the disease. Full-elliptic springs used in connection with the Hotchkiss drive are not greatly open to the above objection, but have the defect of being much more badly adapted to absorb torque-reaction loads than are semi-elliptic springs.

THE CRANE SPRING-SUSPENSION

With the idea of combining the simplicity and other advantages of the Hotchkiss drive with the proper cushioning of the shocks just described, the device illustrated in Figs. 1 and 2 was conceived. In addition, it appears in practice to remove two of the principal disadvantages of the Hotchkiss drive, by greatly increasing the resistance to torque reaction of any given set of springs without entirely removing the desirable cushioning effect to driving and braking shocks and, because of this fact, doing away almost completely with the jumping action of the rear axle on soft roads, already criticized.

Fig. 1, which is a schematic drawing of the arrangement, shows a semi-elliptic spring, shackled at both ends, and rigidly bolted to the axle. It shows also a distance-rod, the forward end of which is connected to the frame by a ball-and-socket joint and the rear end is connected to the axle by a similar joint, not at the center-line of the axle, but at a point substantially below the center-line, a

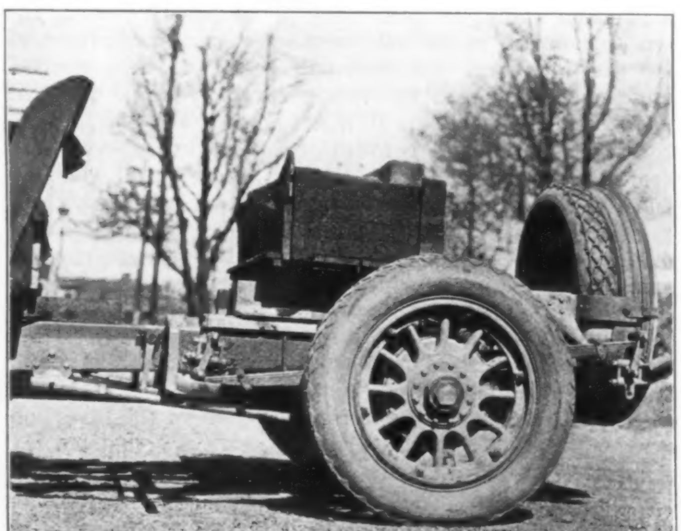


FIG. 2—APPLICATION OF THE SPRING-SUSPENSION ILLUSTRATED IN FIG. 1 TO A MOTOR-TRUCK CHASSIS

bracket rigidly attached to the axle being provided for the purpose. For clearness, only one end of the axle is shown on the drawing, the equipment of the other end being the same.

It is obvious that, with this arrangement, the axle is free to move longitudinally with respect to the frame as well as vertically, while at the same time it is constrained in both these movements by the resistance of the semi-elliptic spring. The vertical resistance need not be explained, being the normal spring action. The resistance to longitudinal movement is occasioned by the fact that, to translate longitudinally, the axle casing must rotate about the ball-and-socket joint located below the center of the axle. This rotation is resisted by a couple set up in the spring, tending to increase the load on one arm of the spring and decrease it on the other. The character of this resistance evidently can be varied by changing the distance of the point of attachment of the distance rod below the center of the axle. The spring shackles are made long enough not to interfere with this action. It is also necessary to provide for extra longitudinal come-and-go in the propeller-shaft connection.

In the conventional Hotchkiss drive, the torque reactions due to driving and braking are resisted by couples set up in the springs. This is still the case in this new arrangement, but the couples are of considerably less magnitude due to the action of the lever-arms to which the distance-rods are attached. The reduced twisting of the axle is very evident in actual driving, as is also the improved action over soft rough roads when using heavy power. The latter improvement undoubtedly is caused by the better method of taking care of the torque reactions and by a different type of tire reaction with this suspension.

Tire Reaction

This question of tire reaction is a very important one where the axle weight is high compared to the total weight of the vehicle. The pneumatic tire, especially one of the cord type, constitutes a most efficient spring. In other words it gives back practically everything that is put into it, with a minimum of damping action. The tossing about of such a weight between the tire spring on one side and the vehicle spring on the other certainly interferes with the smooth riding of the vehicle as a whole. This is why it is desirable to reduce the axle weight to the lowest possible figure. Unfortunately, it is not feasible in the case of the shaft-driven axle, even with the most careful design, to reduce the weight sufficiently to allow of ignoring it altogether. The new method of suspension, just described, is believed to provide a considerable improvement over conventional arrangements by reducing materially the maximum compression in the tire under any given set of operating conditions, and by giving an attachment between the axle and the frame that is cushioned in all directions except laterally.

One cause of excessive tire compression was partially explained in a previous paragraph that outlined the forces acting on an axle rapidly traversing a rough road. This action can be understood most easily by examining the action of an axle crossing a hump on the roadway. If rigid distance-rods are used, the momentum of the axle in a horizontal direction is augmented by that due to the total car-weight to a degree depending on the angle and the height of the hump. The reason for this is that the line of force through the wheel to the wheel hub and so to the axle is inclined at a considerable angle by the action of the hump, although this line is substantially vertical when the wheel is traversing a smooth level surface. The inclined force resolves into vertical and horizontal components at the axle center. The opposing forces are, vertically, the weight of the axle and wheels plus the pressure of the springs and, horizontally, the total weight of the vehicle. At low speeds, the momentum effect of the masses is relatively small. At high speeds, it is very great.

In a previous paragraph I called attention to the fact that the flat semi-elliptic spring is deficient in flexibility except in a vertical direction. This is due to its stiffness as a beam in a lateral direction and to its great torsional stiffness. The use of rebound clips, tying the plates more or less rigidly together, is a contributing factor to the foregoing. If all irregularities of the road surface were symmetrical and were crossed at substantially right angles by the axle and the wheels, the lack of flexibility in directions other than the vertical would be of no importance. Practically, however, the contour of the road surface traversed by the wheels on one side of a car is rarely the same as the contour of the surface traversed by the wheels on the opposite side. This fact in itself tends to cause lateral shocks. These are augmented by the constantly changing relation between the axle and the floor of the car which do not remain parallel but vary from parallelism by considerable angles in either direction. Such changing angular relation can be accomplished only by the springs or frame, or both, flexing both torsionally and laterally. Of course, this action tends to reduce what is commonly called "rolling" on curves and heavily crowned roads, but it does so at the expense of some severe strains and shocks in the parts involved, which produce an unpleasant jarring effect in the riding on cobble stones and similar road surfaces. It is the action just described that makes it so difficult to keep the rattles out of some of our modern passenger-cars. Actually, a very few thousandths of an inch of side-play of a spring in a shackle is sufficient to cause a most unpleasant noise.

Fig. 2 shows one method of improving this action where semi-elliptic springs are used on a passenger-car chassis. The springs are connected to the chassis frame at both ends by short links that allow of practically unlimited twisting and a limited amount of side swing. It is possible that this arrangement may give rise to an excessive amount of rolling, but tests so far made, with the chassis loaded as shown in the photograph, do not indicate this. To minimize the chances of rolling, the springs are carried as high as possible, being mounted above the axle. The springs are tilted down in front, so as to clear the doors and floor of the body. It is probable also that stiffer springs, that is, springs having a steeper scale in pounds per inch of deflection, can be used because of the better all-round cushioning effect obtained. One advantage of the full-elliptic type of spring is its better action in cushioning lateral shocks. If it were not for its weakness in resisting axle torsion, a lack of faith on the part of designers in regard to its ability to position properly the steering axle in heavy-duty service and its interference with low-hung bodies when used for rear-axle suspension, it might well become a popular type for passenger-car use.

The cord pneumatic tire, due to the low air-pressures that can be used successfully with it, has been the greatest single factor in improving the riding of automotive vehicles in recent years. The object of this paper is to call attention to the fact that there is still much that can be improved in the design of the other parts contributing to riding comfort.

Flighty Reflections

By C. L. Egtvedt[1]

Semi-Annual Meeting Paper

PROPHECY having always been a forerunner of great things, we now expect any new and important development to have a prophetic introduction. As the engineer is seldom a "dreamer," his forecasts usually are not true prophecies but merely projections of the past into the future. Notwithstanding the great advance of aviation during the war, we now are beginning to realize that little real progress has been made since the original successes of the Wright brothers and Count Zeppelin, and inquiry is made whether the efforts of inventors are being directed along proper lines.

By comparing the weights and organisms of birds, insects and fish it became evident that their constitution is not essentially unlike that of land animals and that the only difference in their manner of locomotion came by the use of different mechanisms.

All efforts heretofore having been directed toward the perfecting of the stationary wing and high-speed propeller combinations in airplanes, the flight of only a small class of insects and soaring birds can be imitated and the versatility of other types, such as the fly and the humming bird, has not been approached. To obtain this freedom of action, the conclusion is drawn that we should incorporate in heavier-than-air machines all the features that nature demonstrates in the various forms of animal flight. Comparing the development of aircraft service to that of the locomotive and the automobile, the lack of proper landing-fields and aids to navigation is discussed and the question is asked whether we are not nullifying the advantages of air transportation by admitting that the requirements of present-day craft cannot be greatly improved and constructing exceedingly large suburban landing fields, rather than concentrating our efforts on the fundamentals of flight as exemplified in nature.

NO important movement in human affairs has ever come to pass without first having been introduced; and contemporary environment has always been gradually modified by advance pressure of the coming event, so that the arrival has never been a complete surprise. Each decade develops a crop of prophets, whose gift it is to turn their eyes to the future and preach the advent of that which is to come, influencing and molding the respective environment for its most favorable reception or building up reactionary forces to weaken the shock of the tidal wave.

Prophetic utterances are, as everything else in nature, the necessary effect of natural causes and generally these causes are strong demands or pressure arising from gradual changes in environment upon other factors of civilization that have lagged behind the average movement of events or fallen out-of-line too far ahead of the ranks.

The steam engine, the dynamo, wireless communication and the flying machine, at the time of their first reception by the general public, have all given rise to flights of fancy that over or undershot actual final accomplishment, but which at the same time urged the inventive brain to continual effort toward materialization of these prophecies, especially if made by men of accepted standing in the art. However, a broad vision and the ability to forecast are rather exceptional gifts and we are content to classify those who are afflicted with them as "dreamers." So regularly has prophecy been a forerunner of great things that we now expect prophecy before we expect anything new and important in the development of our affairs, and when we launch an enterprise that we hope will be a sensation, prophecy must introduce it. It has become almost a custom at the inauguration of the presiding officers of engineering societies to expect from the new leaders something prophetic. But the engineer is seldom a "dreamer." As a consequence we have had forecasts that are not true prophecies but merely projections of the past into the future. Definite outlines of research usually are given and it is due to these outlines that past art has in many cases been improved, but it is also due to these outlines that inventive effort has been constrained into definite channels. What if the guidance was misleading? What if the past efforts have been along a line not altogether based upon correct and scientifically sound principles?

When the Wright brothers and Count Zeppelin stirred the world by their first successful flights, it began gradually to dawn upon us that man would some day move through the air as easily and safely as a bird. Especially after the great advance immediately preceding and during the war did we expect that the complete conquest of the air was near at hand. Considering all this strenuous development in retrospect, we begin now to realize that we have achieved comparatively little in advance of the original successes of these first inventors. The present-day types of aerial vehicle differ very little from the original designs and we have assumed so far that no fault should be found with this conventional coordination of sustentation and propulsion as exemplified by the fixed wing or gas-bag and the screw propeller respectively.

The First Incentive

Man received his first incentive for aerial locomotion when he recognized that birds or insects do not sustain themselves in the atmosphere by any supernatural agency but pay for their apparent defiance of the law of gravitation by the expenditure of energy. When people began to insist upon an explanation of natural phenomena, originally explained in an orthodox way by Biblical citations, and insisted upon an explanation on the basis of natural laws framed around the principle of conservation of energy, flight as exemplified by nature was scrutinized with increasing interest. It was, in the first place, discovered that the weight of the bird or insect is actually greater than the weight of the air that it displaces; and, secondly, it was found that the organism of these flying beings is not constituted so much differently from that of animals that move on terra firma as to make us suppose that they can store in their muscular tissues much greater amounts of energy in comparison with their weight than those animals. It became evident that the difference in the manner of locomotion came about only by the use of different mechanisms, but coordinated to the properties of the medium, air, so as to produce the desired motion in accordance with, and not in violation of, simple natural laws.

Action being understood to be equal to reaction explains that the force acting on the body of the bird in a

[1] Chief engineer, Boeing Airplane Co., Seattle, Wash.

vertically upward direction must be equal to a force exerted by the body in a downward direction and both these forces are equal to the mass set in motion times the acceleration of that mass. The upward force is equal to the mass of the body times the upward acceleration of the body and of the air that moves with it, and the downward force is equal to the mass of the wings and of the air moved by the wings times the acceleration of wings and air. Upward movement of the body and wings, or flight, could then be brought about only by an unbalance of acceleration and of the masses of air set in motion in favor of greater magnitudes involved in the downward movement of the medium air so as to overcome the force of gravity. Close analysis of the wing reaction upon the air, especially when the upward and downward movements are combined with the horizontal movement, reveals the fact that the air can be deflected in a downward direction during both the downward and the upward movements of the wings, provided the plane of the wing is deflected during its cyclic strokes so that it forms an angle of incidence with the air on its under side at all times. This is accomplished by inclining the wing on the downward stroke so that the leading edge remains at a lower elevation than the trailing edge and vice versa on the upward stroke. The wing motion of the bird is, however, not confined to motion in a vertical plane. In actual flight the tips of the wings describe an ellipse with respect to the axis of the bird body; or a cycloid with respect to their movement in the air. This means that the bird wing is brought forward at the top of the stroke and is again moved toward the rear at the bottom of the stroke. If it be assumed that the bird travels at the rate of 30 m.p.h. and that the wing motion along the path of the ellipse is uniform at the rate of 30 m.p.h., the return movement at the top of the stroke has a velocity greater than the velocity of the body of the bird in the air to the extent of 30 m.p.h.; hence, at the top of the stroke, the velocity of the bird wing in the air is 60 m.p.h. If, for a given airfoil area, the lift is proportional to the square of the velocity, the bird wing on its horizontal movement would have a lifting capacity four times the lift of a wing, which is considered fixed to a moving body, as, for instance, the wing of a bird in soaring or gliding flight, or the wing of an airplane. At the bottom of the stroke, the bird wing travels rearward at a rate of 30 m.p.h. The body of the bird is moving forward through the air at the same speed; hence, the relative motion of the wing in space must be zero. This condition of minimum velocity of the wing with respect to the medium lends itself ideally to the reversal of inclination of the wing chord, which changes, as above stated, from an upward slope from the leading edge to a downward slope from the leading edge during the downward and upward strokes respectively.

Function of the Wing Joint

It is important also to note the function of the wing joint of birds that allows the hinging of the outer extremity of the wing about a horizontal axis. Unlike a screw propeller, the point of maximum efficiency of which is about two-thirds the length of the radius from the hub, with the efficiency becoming less from that point radially inward and outward, the hinged-wing section of the bird can operate with the same efficiency from the hinges to the tip, if the velocities of the relative movements of the wing and the air and the wing profile are constant from the hinge to the tip, with no consideration of tip losses.

The above analysis will tend to indicate that the bird wing can perform the functions of both sustentation and propulsion and will indicate also that, only with great difficulty, after a series of refinements of design yet to be accomplished, would aerial vehicles of the present type approach the efficiency and versatility of movement in all directions that nature shows us are so well accomplished in the movable and flexible wings of the bird.

Insect flight seems to resemble more closely the mechanism of the airplane inasmuch as some classes of insect have separate mechanisms for propulsion and sustentation. The wing covers, usually made up of rigid tissues, are spread out in flight and held stationary while the wing membrane proper operates below these cover planes in rapid rotary motion resembling very nearly the motion of a screw propeller. This high-velocity movement causes impulses upon the air that are frequent enough to give rise to a musical sound.

Another class of insect does not possess the stiff wing-covers and both sustentation and propulsion are accomplished by the propeller action of the wings. These insects exhibit almost unlimited versatility in movements which vary from zero to maximum velocity in almost any direction with respect to the axis of the insect body.

The lighter-than-air machine finds its parallel in nature in the form of the fish. There are no organisms that can operate in the air on the principle of buoyancy, for the reason that we lack sources of lighter-than-air media that could be used against the force of gravity to advantage in vertical movement. The lightest gas so far known has about one-twenty-ninth the weight of air, whereas air has about one eight-hundredth that of water. Apparently nature could not accomplish the producing of aerial fish, a task that would be about $800/29 = 28$ times as difficult as the creation of water fish, even if hydrogen were ever present as a surface layer above our atmosphere. However, man has found a means of drawing rather easily the lighter gases from nature's unlimited resources and with this obstacle overcome has attempted the task of floating in the air in spite of the manifold difficulties of operation to which nature's organisms, fish, are not subjected, as, for instance, landing. It may be noted that flying-fish belong in the heavier-than-air class, being equipped with large pectoral fins that permit them to sustain flight for a limited period.

Up to the present time practically all efforts in aeronautics have been directed toward the perfecting of the stationary wing and high-speed propeller combinations, the airplane being capable of imitating the flight of only a small class of insects and the soaring flight of birds when the propeller is inactive. Attempts have been made in the helicopter to simulate the flight of insects of a more versatile type, which depends upon propellers only for sustentation and propulsion; but great difficulty has been experienced in providing sufficient control for the purpose of directing the propeller thrust in other than a vertical direction. It also appears that to depend upon sustentation by propeller action alone takes away the feature of safety, a feature that is never entirely realized in our complex combustion engines so much as it would be in a fixed wing truss composed only of parts at rest with respect to one another.

Importance of Versatility

It appears that with the development of the airplane only a small step has been made in the way of accomplishing satisfactory flight. We should incorporate in our heavier-than-air machine all the features that nature demonstrates in her various forms of animal flight to give us the same feeling of being perfectly at home near

the clouds that we observe in the graceful and playful motion of the birds or the dancing flight of the fly, or of the humming-bird hovering in the atmosphere, apparently without effort, darting to the right or the left or forward with a rapidity and ease that will be envied by the human aeronaut until he can do the same.

It further appears that we have been trying to perform successful commercial operations without questioning to any great extent our present operating mechanisms. Sustentation has been provided by forward motion except in the case of lighter-than-air craft in which sustentation is obtained by principles of buoyancy alone, or in combination with a forward movement. In this manner we have accomplished sustentation but with an apparent lack of versatility in the application of energy. Instead of being able to sacrifice propulsion to aid sustentation at the lower speeds and to sacrifice sustentation to aid propulsion at the higher speeds, we find our propelling force decreasing with the increase of the speed of advance and our sustaining surfaces greater than we can efficiently use at these higher speeds. Are natural flight requirements solved in this manner, or does nature utilize energy with greater discretion? Although the history of flight does not show that a flexible propelling mechanism has ever been operated successfully, we are not in a position to say that this means of propulsion may not be accomplished in a manner that will advance materially our efforts to produce satisfactory flight.

When considering flight from the point of view of those engaged in the construction of aircraft, a number of questions arise which indicate that the art must advance before stability can be expected. Can we truthfully say that the actively thinking public is satisfied with flight as exemplified by the past and present art, or are we still confronted with the problem of solving flight in a manner that will produce absolute confidence and provide genuine service? If the former were correct, we should have little difficulty in constructing and marketing our products. However, if the latter is correct, shall we try to force the building of existing types or develop new aircraft art?

Recalling some of the principal methods of transportation, we note that it was approximately 50 years after the invention of the locomotive before the service offered was acceptable to the extent of making the railroads pay. More than 20 years of development was necessary to provide the service the general public required from the automobile. We believe the public of today has a greater knowledge and appreciation of mechanical progress and is learning to accept it more readily; yet, if we realize that mechanical flight was accomplished in 1903, we must admit that willingness to accept is prompted by some other reason than a new public viewpoint. The general public has learned to invest in service and only a few persons are willing to invest in limited service. In a spirit of desire to provide service, we attempt to analyze the present conditions that indicate limited service and, turning to aircraft operation, we find our present types hampered by the lack of proper landing-fields and aids to navigation. It is true that efforts expended in this direction would greatly improve the service now obtainable, but are the requirements for landing-fields reasonable, or are we partially nullifying the advantages of air transportation by the necessary construction of exceedingly large suburban landing-fields? Are we satisfied that the requirements of our present craft cannot be very greatly improved, or are we planning to concentrate our efforts on fundamentals of flight to approach more nearly the flight exemplified in nature?

We have constructed in our aircraft advance the airplane, the airship, the helicopter, the ornithopter and the free balloon; and have improved them in varying degrees until it appears that it is very easy for us to continue these developments. So long as these developments indicate reasonable improvement, we are not likely to question the fundamentals of our work or attempt widely different fields of exploration unless forced to do so. We have advanced and are progressing at present, but the progress at present is largely the same as that of the general automotive industries; and if we were to analyze carefully the development of aircraft since controlled flight was first accomplished, would we find the greatest advancement in the methods of sustentation and propulsion, or in the coordination of structural units and the employment of improved structural materials and advanced methods of construction?

WELFARE WORK

WE are trying to ameliorate the prejudice that has grown up between employer and employe. We want them to feel better toward each other than they have in the past. One way is to teach the employe that the employer cares for him. That is one function of the welfare work. It is to instil in the minds of working men that they are not working machines, that they are not to be scrapped like iron and steel, but that they are human beings and have human souls. What does God Almighty care whether a man is worth $1,000,000 or 10 cents? What we are trying to teach is brotherhood. We are trying to instil in the minds of employers that they have duties toward their employes; that the mere possession of capital is a trust; that the carrying on of business is a trust, and that the human beings they employ are entitled to be considered as brothers. But in teaching this we are also teaching the working men that they should be brothers to each other, and learn to regard their employers as friends.

What would you do if you were a day laborer and all there was for you on earth was to work hard all day, go home tired, get your dinner and go to bed, only to waken the next morning to repeat that routine day after day for the remainder of your life? Then what would you say if you were a little higher in the scale, a mechanic, for example, and had to make the same things, work on the same piece of metal day in and day out; that all you had to look forward to was getting up in the morning and doing something you did not care to do, doing it all day, and all the change you had from that mechanical routine was the little social enjoyment you had with your family and your children and going to the movies? How would you feel under those circumstances? Then let us go a step higher. Suppose you were a banker and all the work for you to do was to go to your office and scheme out how to get the best of some other man; how to make money for yourself and your bank, and to get engagements to float securities and then how to sell? Suppose you were scheming to do that all the time, and every day had that routine of collecting and disbursing money. What would your life be? There are many sterile souls among the leading men of this country, many among the commercial men, among the mechanics and among the laboring people.—Haley Fiske.

Airplane Performance Formulas

By Edward P. Warner[1]

Semi-Annual Meeting Paper *Illustrated with* Charts

AERODYNAMIC analysis relates mainly to questions of performance and stability, the latter including both maneuverability and control, but the designer's problems concern chiefly the prediction of the best possible performance. Accurate analysis, which would include a summation of the elemental resistances of an aircraft part by part and the making of many corrections, supplemented by tests of models in a wind-tunnel, involves much labor and expense.

When a preliminary choice of dimensions and specifications for a new type of an airplane is to be made or there is a question of the performance attainable with a given load and power, a shorter method becomes necessary. This is to be found in the derivation of simplified formulas and graphs.

The author illustrates by examples the process of deriving these formulas and considers in turn such elements as minimum and maximum speeds; climbing ability and the conditions under which an airplane will have a zero ceiling, that is, the limiting conditions under which flight is possible; rate and angle of climb, the latter controlling the possibility of getting out of a small field over a barrier; and the best ceiling possible. Under these heads he takes up such questions as fineness, which is the ratio of the parasite resistance to the wing area; and lift, weight and power and their relations in determining the various coefficients which are used. Values obtained theoretically from the formulas were checked by comparison with those of about 50 airplanes of various types of which the measurements and performance are known and application is made to specific examples. Numerous curves show the relation of the various coefficients that enter into the design and performance of the airplane.

THE aerodynamic analysis of airplanes and that portion of airplane design dealing with the practical applications of aerodynamic theory divide naturally under the two heads of performance and stability, stability being taken to include maneuverability and control. A large share of the designer's problems, therefore, pertains to the prediction of performance and to the determination of a design that will give the best possible performance.

The analysis of performance may be carried out by several methods, the more refined and accurate of which involve much labor. To obtain a really accurate estimate of the speed or climb of an aircraft necessitates the summation of the elemental resistances part by part and the making of many corrections for such factors as the slipstream, the inclination of the thrust axis, interference between the parts of the structure, and the like. The labor involved, while justified in a final calculation, is too great to be undertaken lightly when preliminary estimates of performance are to be made or when a number of different possible arrangements of the parts of a machine are to be compared with respect to their aerodynamic qualities.

A much simpler device for approximating performance involves the use of a wind-tunnel test of a complete model. This is less accurate than the method of direct calculation, as the wind-tunnel model is so small that it is impossible to make it a true scale reproduction of a full-sized machine in every particular. A multitude of such items as fittings and stranded wires, small individually but of importance in the mass, are therefore omitted from consideration. Even if it were possible to include such parts, the scale effect would be so enormous that the results obtained by the usual method of scaling in proportion to the square of a linear dimension and to the square of the speed would give incorrect results. A wing made to 1/20 scale may logically be expected to give very nearly 1/400 the lift and drag of the full-sized wing at the same speed, but the results that would be obtained by attempting to make a 1/20-in. scale model of a cable 1/8 in. in diameter or of a honeycomb radiator would be almost meaningless and certainly would not repay the vast amount of effort involved. Furthermore, the wind-tunnel does not allow for the making of a correction for slipstream effect, a very important factor in the resistance of an airplane. On the whole, it is rather remarkable that notwithstanding these obvious and manifold disadvantages it is still possible to secure a reasonably close approximation to the performance of an airplane by a single test of the lift and drag of a complete model in the wind-tunnel. The results for maximum speed are somewhat better than are those dealing with climb, as the slipstream effect, ignored in the wind-tunnel test, is of relatively more importance with reduced air-speed and open throttle, which is a climbing condition, than at maximum speed in level flight.

The wind-tunnel is very valuable and it is difficult to imagine where we would find ourselves today in aeronautical engineering if no tunnel tests of complete models had ever been made. Certainly our knowledge of stability, as well as our ability to predict performance, would be on a far lower plane than they have actually attained. Entirely aside from any discussion as to the theoretical merits of wind-tunnel testing for performance, however, it is evident from practical considerations that such tests cannot be made in all cases. Lack of time and money often forbids the making and testing of a wind-tunnel model, just as it often prevents the carrying out of elaborate performance calculations.

When a new type of airplane is to be designed and when the engineer is confronted with the necessity of making a preliminary choice of its general dimensions and specification, or when there is put up to him suddenly the question of the performance attainable with a given load and power, there is no opportunity to refer the problem to the wind-tunnel or to cover reams of paper with tabulations of resistance. Some shorter method becomes necessary and that shorter method can be found, assuming that the designer's experience with similar types is not sufficient to enable him to guess the performance offhand, only in the derivation and use of simplified

[1] M.S.A.E. — Associate professor of aeronautical engineering, Massachusetts Institute of Technology, Cambridge, Mass.

performance formulas or of graphs that represent formulas too complex to be readily expressed in analytical form.

Classification of Performance Formulas

Performance formulas may be divided into three principal groups, (*a*) those based solely on experience with airplanes previously constructed, (*b*) those based on rational considerations alone, or perhaps combined with the data drawn from wind-tunnel tests, and (*c*) those which depend on a combination of theory and practice. Up to the present time, the first-named class of formulas has held the premier place and has often proved of great value. These formulas have dealt with the maximum and minimum speeds, with the rate of climb and with

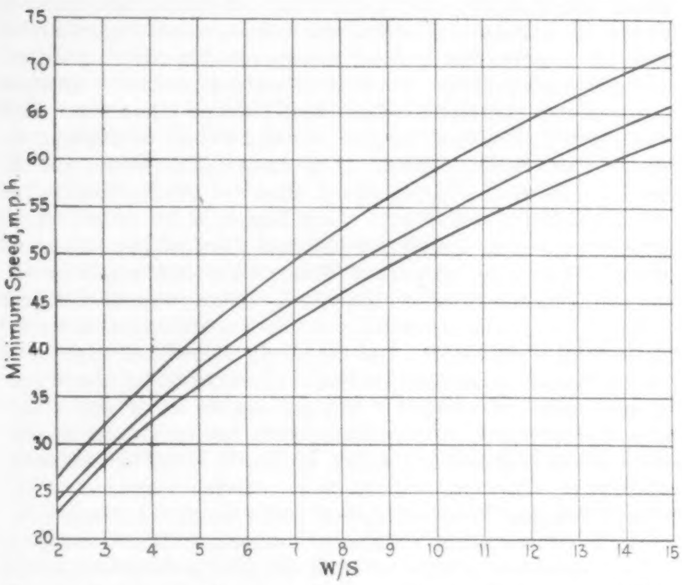


Fig. 1—Curves of Minimum Speed as a Function of the Wing Load

ceiling and with every other conceivable element of performance. They have been no less diverse in their origins than in their purposes. Nevertheless, it is difficult to fit a purely empirical chart or formula to the solution of a problem so complex as that of the flight of an airplane in such a way as to be sure that allowance is made for all the variables that may affect the solution. Only by the use of a modicum of theory can we feel certain that we have omitted from consideration in our formulas no variables that should be included and have included none that are unimportant. This does not imply that pure theory furnishes a satisfactory basis for the derivation of formulas, as it cannot do so in the present state of our knowledge. It is always necessary to depend at some point of the reasoning on experience either in the wind-tunnel or in free flight, but theory has great use in guiding our research and of reducing the probability of going astray in the quest for data. The acid test of any formula is application to existing machines for which the true performances have been determined.

The nature of the formulas that can be secured by a combination of rational derivation and of experimental data can best be illustrated by actually following through the process of deriving such expressions. By taking up the elements of performance one by one and determining first the form into which formulas should be cast and the variables involved, actual numerical expressions are obtained which give, as far as possible, the performance of an airplane in terms of its well-known dimensions and other characteristics.

Minimum and Maximum Speeds

The first and the simplest element of performance to be treated is minimum speed. In virtually all instances the true minimum speed of an airplane in steady flight is determined by the wing-loading and the magnitude of the maximum-lift coefficient. The only exception to this rule is found in the case of airplanes whose reserve of power is so small that the horsepower required to maintain steady horizontal flight at sea level and at the angle of maximum lift is greater than the power available under those conditions, and whose ceiling is so low that it could only be regarded as a freak unsuitable for serious consideration. It should be borne in mind, although it is perhaps hardly necessary to emphasize the fact, that the minimum speed that is calculated from the maximum-lift coefficient and the wing loading is the true minimum, and not the minimum speed that is often reported in performance tests as being obtained by closing the throttle until the airplane is no longer able to maintain level flight. The speed of minimum power is far from being the true minimum speed, the latter being determinable in level flight only by keeping the throttle well open and stalling the machine to such an angle that it is barely able to fly level but staggers along at an angle of attack of 18 deg. or thereabout. Few pilots have ever made a serious attempt actually to reach or to maintain the true minimum speed of flight.

If it be accepted that maximum-lift coefficient and load per unit area are the only critical factors in minimum speed, it should be easy to derive a formula in terms of these quantities. Writing the fundamental equation of lift

$$L = L_c S V^2$$

where

S = the wing area
V = the speed of flight
L = the total lift

the equation can be given a form applicable to this special problem by taking the particular values V_{min} and $L_{c\ max}$ for V and L_c respectively and by substituting W, the weight of the airplane, for L, the lift being equal to the weight in steady horizontal flight. If this be done, the equation of minimum speed takes the form

$$V_{min} = \sqrt{(W/L_{c\ max}\ S)} = \sqrt{(1/L_{c\ max})} \times \sqrt{(W/S)}$$

It remains then only to assume a value of $L_{c\ max}$. If V is given in miles per hour, the values of $L_{c\ max}$ range from 0.0026 to 0.0045. On substantially all modern wing-sections, however, with the exception of a few thick sections used for cantilever wings only, the value will be found to lie between 0.0029 and 0.0038, the higher values appertaining to thick sections and to those with strongly concave lower surfaces. Solving for $\sqrt{(1/L_{c\ max})}$ we see that

$$\sqrt{(1/0.0029)} = 18.6 \text{ and } \sqrt{(1/0.0038)} = 16.2$$

The formula for minimum speed is thus found to be

$$V_{min} = K_1 \sqrt{(W/S)}$$

where K_1 is to be chosen somewhere between 16.2 and 18.6, depending on the general type of wing-section employed. If nothing is known about the wing-section to be used, a rough but nevertheless useful approximation to the landing speed is given by the formula

$$V_{min} = 17 \sqrt{(W/S)}$$

In Fig. 1 curves of minimum speed as a function of W/S are plotted for the maximum, minimum, and mean values of K_1. Claims of landing speeds materially lower than

those shown by the curves should be viewed with strong suspicion.

The maximum speed offers more complex problems than the minimum, as not only the weight and area and the properties of the wing-section but also the power and the parasite resistance are involved. In attempting the derivation of a rational expression the most convenient and most reasonable simple assumption is that the angle of attack at maximum speed corresponds to the angle of minimum-drag coefficient of the wing. Wind-tunnel tests and free-flight work unite in showing this assumption to be close to the truth in most cases. A small change in the angle of attack at the maximum speed, however, has little effect, as the drag-coefficient curve is ordinarily very flat in the neighborhood of the minimum drag. If this assumption be true it is evident that a small change in the angle will have a negligible effect on the drag coefficient, which can then be considered as keeping a constant value. If the drag coefficient of the wings is considered as remaining constant, the coefficient of total resistance may also be taken as a constant, since the change of the parasite resistance coefficient with the angle of attack is obviously small. The equation of the power required at maximum speed may then be written

$$P = (D_c S + R_c s)\ (V^3_{max}/375)$$

where

D_c = the coefficient of wing-drag
R_c = the coefficient of parasite resistance
S = the wing area
s = the parasite resistance area
V_{max} = the maximum horizontal speed in miles per hour

both of these being assumed to remain constant. The power required for level flight at the maximum speed is, of course, equal to the power available, which is the product of engine power and propeller efficiency. Taking the propeller efficiency, which is denoted by η, also as invariable, the equation can be transposed into the form

$$V_{max} = \sqrt[3]{[(P_o\eta) \div (D_c S + R_c s)]}$$

If the ratio of parasite-resistance surface to wing area then be taken as a constant, that is, if the fineness be assumed the same in all cases, the equation becomes

$$V_{max} = \sqrt[3]{[375\eta \div (D_c + R_c[s/S])]} \times \sqrt[3]{(P_o/S)} = K_2 \sqrt[3]{(P_o/S)}$$

The rather surprising conclusion is thus reached that, given a certain fineness of design, the maximum speed of an airplane is independent of its weight, provided the machine flies at its maximum speed very close to the angle of minimum drag of the wing. This condition, as already noted, is realized in high-speed airplanes and the few comparative tests made on such machines with different loadings actually show the speed at sea level to be practically unaffected by small changes in weight.

It would, of course, be possible to find the value of K_2, as that of K_1 was determined, by assuming values of D_c, R_c, and s/S and solving directly for the constant, but it is easier and more accurate to call in the assistance of free-flight tests at this point, plotting the values of P/S and of V_{max} against each other in such a way as to determine the value of the constant. This can be done most conveniently by plotting V_{max} against P/S on logarithmic paper, as the curve obtained in that way serves not only to determine the value of K_2 but also to check or to disprove the deduction that the speed varies as the cube root of the power-area ratio. The characteristics of 65 machines have been plotted in this way in Fig. 2 and the best mean straight line drawn through the group. It is found that, when the slope of the line on the logarithmic chart is measured, the slope is not truly 1/3 but 0.39. The best average straight line corresponds to the equation

$$V_{max} = 127\ (P/s)^{0.39}$$

The two lines parallel to that of best average denote speeds 10 per cent higher and 10 per cent lower, respectively, than those given by the formula. The maximum departures from the formula were found to be 12 per cent high and 17 per cent low for land planes, while the speeds of flying-boats, which are denoted by crosses in Fig. 2, uniformly fell below the speeds calculated by the formula, the maximum discrepancy being 23 per cent. This was to be expected, as the fineness of flying-boats is in general inferior to that of land planes.

The average difference between the speeds calculated by the formula and those found on actual test was 6.2 per cent for all the machines and 5.4 per cent for the land planes alone. These errors may seem large, but the chart includes points for airplanes as diverse in their characteristics as the old JN4D, the Air Service Messenger, the commercial Junkers, the Verville-Packard racer, and the Trans-Atlantic NC boat. In view of the extraordinary range of fineness represented among the machines used, the maximum departure from the mean curve is not excessive. The average errors just mentioned are a good measure of the probable error in using the formula or curve if no attempt is made to allow for differences in fineness. The probable error can be much reduced, however, if some attempt is made to estimate the fineness of an airplane and to modify the constant in the formula accordingly. It is obvious by a glance at

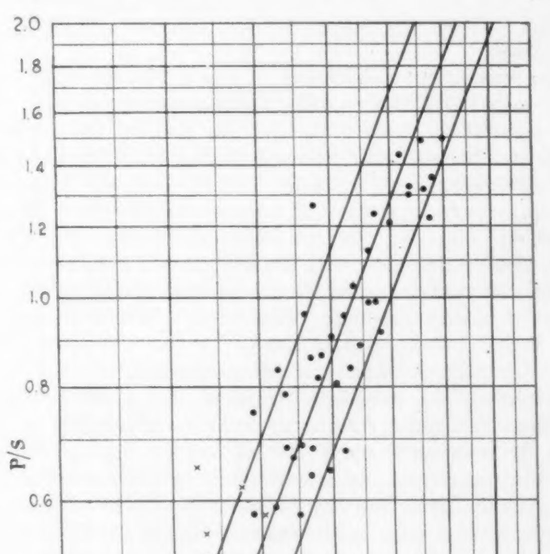


FIG. 2—CURVES SHOWING THE CHARACTERISTICS OF 65 MACHINES AS REGARDS THE RELATION BETWEEN THE POWER AND MAXIMUM SPEED

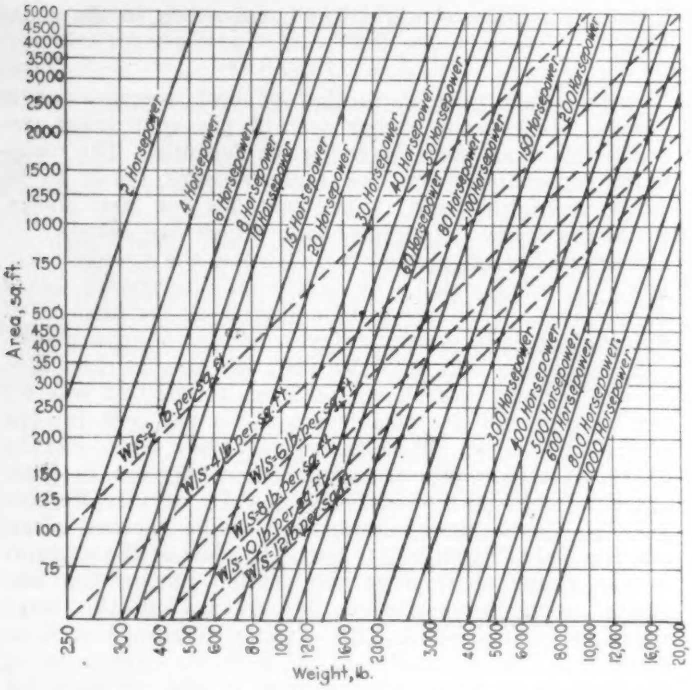


FIG. 3—CURVES OF THE MINIMUM POWER NECESSARY FOR FLIGHT PLOTTED IN TERMS OF THE WEIGHT AND THE AREA

some airplanes that they have more parasite resistance than the average for their type, while others have correspondingly less. The exercise of a little judgment in this respect materially decreases the amount of error liable to appear.

The reason for the choice of the exponent 0.39 in place of $\frac{1}{3}$ calls for a few words of explanation, as this change may appear to invalidate all the theory that has been given and to reduce the whole matter once more to guesswork and empiricism. The variation of the exponent is, however, very logical and indeed might have been expected in view of the assumptions made. In the first place, the assumption was made that the airplanes treated by the formula were all to fly at maximum speed at the angle of minimum-drag coefficient. While it has been seen that this is substantially in accordance with the facts for pursuit and racing airplanes, it is far from representing the true conditions of flight for commercial and heavily loaded bombing craft. Machines of these types fly at maximum speed at angles higher than that of minimum drag, and the wing-drag coefficient is therefore greater than for the small fast types. It is logical to expect then that a decreasing value of P/S will cut down the maximum speed more rapidly than would be indicated by a theory based on minimum-drag coefficients in all cases. Furthermore, the derivation of a single formula for the maximum speed rested on the implicit assumption of equal fineness for all machines. This, of course, is far from the truth and it is found in general that the best fineness, or the lowest values of $R_C s/D_C S$, occur in fast machines having high values of P/S. This again would indicate the probability of such an increase in the exponent of P/S as has actually been found.

A number of other empirical and semi-empirical formulas have been derived for the purpose of finding maximum speed, and a few may be cited for comparison with that just discussed. Bairstow, for example, gives the expression

$$(V_{max} - 55)^2 = 1445\ [(40 \div [W/P] - 1)]$$

for a constant wing-loading of 7 lb. per sq. ft., and the report of the British Advisory Committee on load factors for aircraft presents the alternative formula of similar type

$$([V_{max} \div \sqrt{(W/S)}] - 25)^2 = 196([84.8 \div ([W/P]\sqrt{[W/S]})] - 1)$$

This applies only to single-engine land planes, the values of the constants being changed for other types. Comparison of these two formulas shows a remarkable agreement in view of the modification of the constant, the results being the same within 1 per cent for all reasonable power loadings. Both the British formulas give maximum speeds from 7 to 10 per cent higher than that which I have proposed in this paper. The general agreement in the nature of the change of maximum speed with power loading and wing loading is, however, good; the difference between the results from the Advisory Committee's formula and those from the one derived here being nearly uniform throughout as can be seen from Table 1.

TABLE 1—COMPARISON OF AIRPLANE SPEEDS CALCULATED BY DIFFERENT FORMULAS

		Maximum Speed by Formula, m.p.h.		
W/P	W/S	Bairstow	British Advisory Committee	Warner
7	7	137	136	127
8	7	131	130	121
10	7	120	120	111
12	7	113	113	103
15	7	104	104	95
20	7	93	94	85
9	9	..	136	127
6	6	..	132	127

An American aeronautical engineer has recently devised a maximum-speed formula which has not yet been published, and which I am not at liberty to quote, but which appears to give better results than any of the others that have been mentioned.

CLIMBING CALCULATIONS

After minimum and maximum speed, the most interesting application of formulas is in connection with climb calculations, particularly in the determination of the conditions under which an airplane will have a zero ceiling or, in other words, the limiting conditions under which flight is possible.

The first step in the determination of any climb formula must be the securing of information about the dependence of the minimum power required on the characteristics of the airplane, since all climb calculations depend on the amount of power required for horizontal flight. Such a relation can be obtained by making certain approximations as to the speed at which the best climb is to be secured and as to the L/D of the airplane under climbing conditions. If the L/D is known, the resistance in steady flight can obviously be obtained from the formula $R = W \div (L/D)$ and, if the air speed for the best climb be assumed to be given by the relation $V_1 = K_{min}$, the minimum power required for flight is evidently equal to $WKV_{min} \div (375\,L/D)$, V_{min} being given in miles per hour. Substituting the expression already obtained for V_{min}, this formula becomes

$$P_{req} = [KW \times K_1 \times \sqrt{(W/S)}] \div [375(L/D)]$$

The horsepower available is of course equal to $P_0 \eta$, and it is therefore necessary only to choose values for K, K_1, L/D, and η in order that formulas for the approximate solution of all climb problems can be obtained. In particular, it is evident that the limiting condition under which flight is possible is that in which the horsepower available and the horsepower required are equal at sea level, or that in which

$$W/P \times \sqrt{(W/S)} = [375(L/D)\eta] \div K_1 K$$

At the angle of best climb, wind-tunnel tests and such free-flight information as is available alike indicate that the L/D for a complete airplane may be expected to lie in the neighborhood of 7.5. It has already been seen that 17 is the average value of K_1. The propeller efficiency under climbing conditions may be taken as 70 per cent. Finally, experience shows the climbing speed of an airplane to be on the average 35 per cent greater than the minimum speed, and K may thus be taken as 1.35. Combining these factors, the equation for horsepower required becomes

$$P_{req} = 0.0082\, W \times \sqrt{(W/S)}$$

and the limit of possible flight is given by the condition

$$(W/P) \times \sqrt{(W/S)} < 85.9$$

The minimum power necessary for flight in accordance with this formula is plotted in Fig. 3 in terms of weight and area. The full lines in Fig. 3 represent various engine powers, while the dotted lines are loadings in pounds per square foot. The application of the chart is simple. If, for example, it be desired to find the minimum power with which a total load of 3000 lb. can be carried without going to a wing loading of less than 4 lb. per sq. ft., the power is read off at the point corresponding to 750 sq. ft. and 3000 lb. and found to be 69 hp. If a reserve of 25 per cent of the total horsepower is desired as a minimum, it is deduced that the minimum practical power with which such a machine could be taken off the ground would be 69 x 4/3 or 92 hp. Similarly, if we seek to find the maximum load that can be carried by a 100-hp. engine with a wing loading of not less than 6 lb. per sq. ft., the result will be found at the intersection of the full line marked 100 and the dotted line representing 6 lb. per sq. ft. This intersection corresponds to 3500 lb. and 585 sq. ft.

The rate of climb is determined by the familiar method

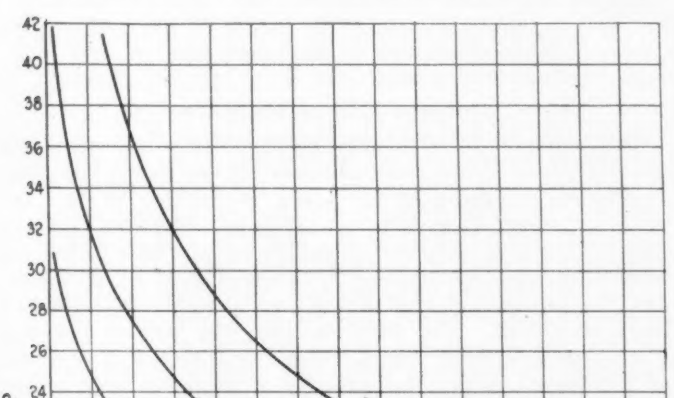


FIG. 4—CURVES SHOWING HOW THE RATE OF CLIMB AT SEA LEVEL SHOULD VARY AS A FUNCTION OF THE LOADING PER SQUARE FOOT AND THE LOADING PER HORSEPOWER

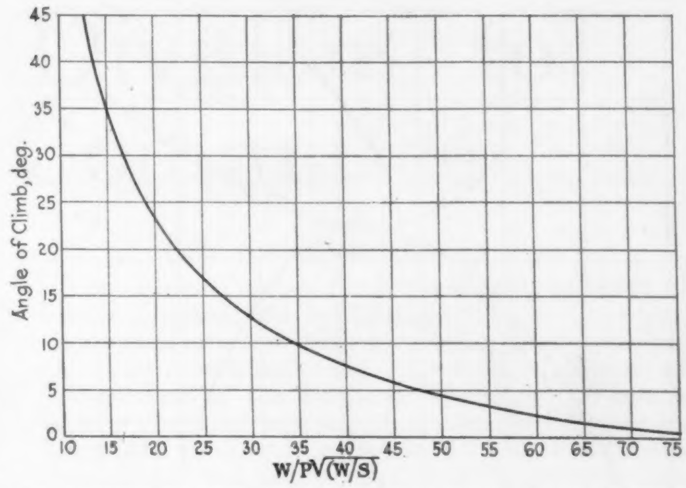


FIG. 5—RELATION BETWEEN THE ANGLE OF CLIMB AND THE WEIGHT

of dividing the excess of power available over the power required by the weight of the machine. The rate of climb in feet per minute at sea level should therefore be given by the formula

$$V_c = 33{,}000\,[(P/W)\eta - 0.0082\sqrt{(W/S)}]$$

In this connection it will be wiser to take 65 per cent rather than 70 per cent as the propeller efficiency in climbing, since the machine that is barely able to fly will have its propeller designed for the one speed at which it can get off the ground, while the airplane with a more normal climb and reserve of power has its propeller designed for a speed nearer the maximum and therefore works more inefficiently under climbing conditions. Taking account of this the expression becomes

$$V_c = [(21{,}400 \div [W/P]) - 271\sqrt{(W/S)}]$$
$$= [271 \div (W/P)] \times [79 - ([W/P] \times \sqrt{[W/S]})]$$

Fig. 4 shows how the rate of climb at sea level should vary as a function of the loading per square foot and the loading per horsepower.

Another performance factor of interest, especially in commercial airplanes, is the angle of climb, which controls the possibility of getting out of a small field over a barrier. The sine of the angle of climb is, of course, proportional to the ratio of the climbing speed to the air speed. The formula for the climbing speed is given just above and it has been seen that the air speed in climb may be taken as 22.9 $\sqrt{(W/S)}$ in miles per hour, or 2015 $\sqrt{(W/S)}$ in feet per minute. The formula for climbing angle therefore becomes

$$\sin\theta = (10.62 \div [(W/P)\sqrt{(W/S)}] - 0.135)$$

A curve of θ against $(W/P)\sqrt{(W/S)}$ is plotted in Fig. 5.

CEILING FORMULA

The determination of the formula for ceiling in terms of weight, area and power alone requires some preliminary analysis in order that the ceiling may be found in terms of the horsepower available and the horsepower required at sea level. It is well known that the minimum horsepower required for flight varies inversely as the square root of the air density, and it has been found by experience that if an exponential relation is required a satisfactory approximation to the variation of the engine power with the altitude is given for most engines by the formula $P = P_o^{1.15}$, P_o being the power at sea level. The maximum power available and the minimum power required must be equal at the ceiling if the propeller is designed so as to give the best ceiling possible.

The horsepower curves at the ceiling with such a propeller would have a common horizontal tangent, as indi-

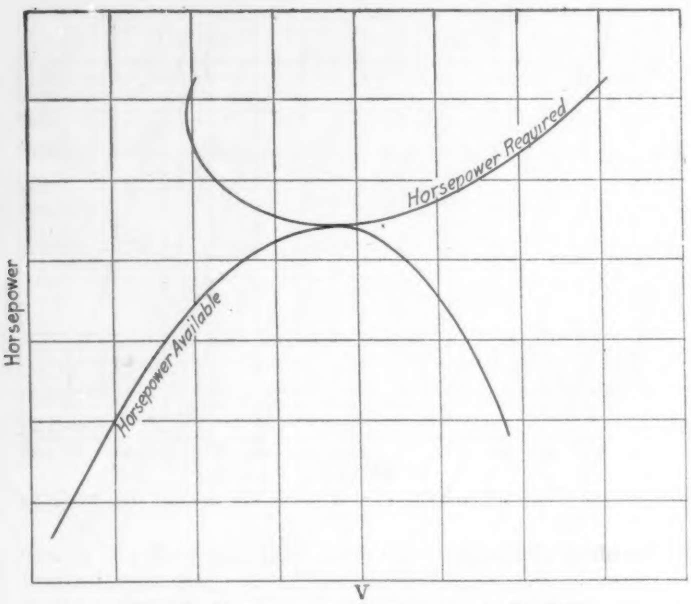


FIG. 6—HORSEPOWER CURVES AT THE CEILING

cated in Fig. 6. The condition of flight at the ceiling would then be expressed by the equation

$$P_{ro}(\rho_h/\rho_o)^{-0.5} = P_{ao}(\rho_h/\rho_o)^{1.15}$$

or

$$(\rho_o/\rho_h)^{1.65} = (P_{ao}/P_{ro})$$

where

ρ_h = the air density at the ceiling
ρ_o = the air density at sea level
P_{ro} = the minimum horsepower required at sea level
P_{ao} = the maximum horsepower available at sea level

Taking the logarithms of both sides of this equation we have

$$1.65 \log(\rho_o/\rho_h) = \log(P_{ao}/P_{ro})$$

The relation between the air density and the altitude under normal temperature-gradient conditions is given with sufficient exactness for most practical purposes by the formula

$$h = 69{,}000 \log_{10}(\rho_o/\rho_h)$$

Substituting in the power-density equation the value for the logarithm of the density ratio obtained from this density-altitude relation the formula for ceiling height becomes

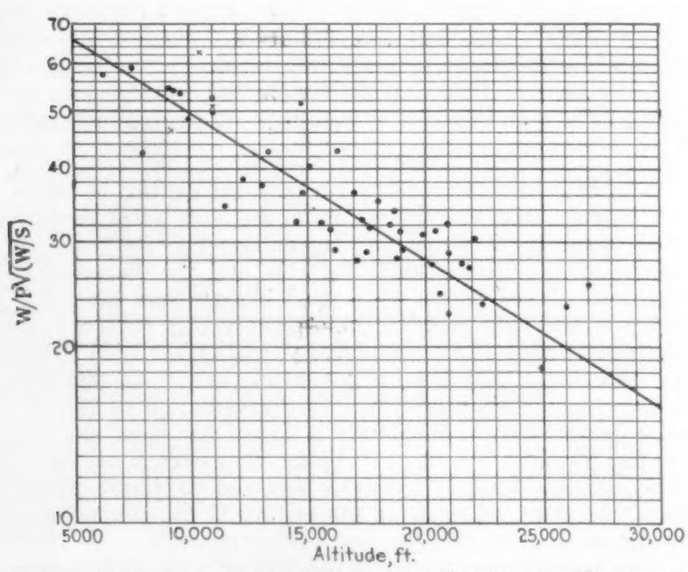


FIG. 7—APPLICATION OF THE FORMULA FOR CEILING TO A NUMBER OF AIRPLANES

$$H = 42{,}000 \log_{10}(P_{ao}/P_{ro})$$

Referring back to the approximate values given for P_{ao} and P_{ro}, however, it will be seen that their ratio is equal to $85.9 \div (W/P)\sqrt{(W/S)}$ and the theoretical formula for absolute ceiling in terms of the principal characteristics of the airplane therefore appears to be

$$H = 42{,}000 \log_{10}[85.9 \div (W/P)\sqrt{(W/S)}]$$

This formula, like the one for maximum speed, has been put to the test of application in a number of airplanes for which performance data were at hand, and the results are shown in Fig. 7. Semi-logarithmic paper was used in this instance in order that the formula just given might plot as a straight line. The points are so grouped as to leave no doubt that absolute ceiling is primarily a function of $(W/P)\sqrt{(W/S)}$, just as maximum speed has been seen to be primarily a function of P/S. The process of drawing the best average straight line through the points in Fig. 7, however, indicates that the ceiling formula, like that for maximum speed, can be improved by slight changes in the values of the constants, the changes in the present instance being very small. The actual form of the expression giving the best mean results for the 46 airplanes considered proves to be

$$H = 40{,}000 \log_{10}[87.6 \div (W/P)\sqrt{(W/S)}]$$

The average error in predicting the ceiling by this formula was 12.6 per cent. Absolute ceiling is both harder to calculate and harder to measnre than maximum speed, and it is therefore natural that the average error in the application of the formula should be somewhat larger for the ceiling than for the velocity.

If the time to climb to a particular altitude is required it can be found by calculating the absolute ceiling and the rate of climb at sea level and then applying the familiar climb formula

$$t = -(H/V_{co}) \cdot \log_e[1 - h/H)]$$

where

H = the absolute ceiling
t = the time to climb to the height h
V_{co} = the initial rate of climb

APPLICATIONS OF FORMULAS

In closing, it is of interest to consider a few examples showing how these formulas and curves can be applied to specific problems in the quick estimation of the approximate performance of an airplane. Suppose, for example, that it is required to design an airplane to have a maximum speed of 100 m.p.h. to carry 4000 lb. of useful load, including fuel, and to reach an absolute ceiling of at least 12,000 ft. The total weight, the power and the wing area are to be found. The requirement of a speed of 100 m.p.h. gives immediately a power-to-area ratio. By reference to Fig. 2, this is found to be 0.54. A similar reference to the ceiling chart, Fig. 7, shows that an absolute ceiling of 12,000 ft. requires that the value of $(W/P)\sqrt{(W/S)}$ shall not exceed 44. It is now possible to solve directly for the power loading and the wing loading, which are found to be 15.3 lb. per hp. and 8.3 lb. per sq. ft. Assuming the useful load to be 40 per cent of the total weight, the required quantities are found to be 10,000 lb., 655 hp. and 1210 sq. ft. The approximate minimum speed would be 40 m.p.h.

As a second illustration, we may find the minimum wing-area with which it will be possible to carry a total load of 4000 lb. to a height of 20,000 ft., using a 400-hp. engine. The power loading is then 10 lb. per hp., and Fig. 7 shows that $(W/P)\sqrt{(W/S)}$ must not exceed 27.7 if 20,000 ft. is to be reached. W/S therefore must be less than 7.7 lb. per sq. ft., and the wing area must be at least 520 sq. ft. The maximum speed should be 115 m.p.h. and the minimum, 47 m.p.h.

Reports of Divisions to Standards Committee

THE following Division Reports will be presented at the Standards Committee Meeting, Tuesday, June 20, beginning promptly at 10:30 a. m., at the White Hotel, White Sulphur Springs, W. Va. They are published in this issue of THE JOURNAL sufficiently in advance of the meeting to permit a careful review of the recommendations and the preparation of the discussion by the members of the Standards Committee and others.

The reports of the Divisions will be presented at the Standards Committee Meeting in the order in which they appear hereinafter. Discussion of the reports is invited from all who are technically familiar with the several subjects.

Under the Standards Committee Regulations only members of the Standards Committee may participate in the voting that will follow the discussion of each recommendation at the Standards Committee Meeting. At the Society Meeting only voting Members of the Society may participate in the voting.

All subjects passed at the Standards Committee and the Society Meetings will be reported in the August issue of THE JOURNAL and submitted to a letter ballot of all of the voting Members of the Society, unless contrary direction is given by the Council.

The several Divisions submitting reports, and the Subdivisions that in many instances prepared them, have considered carefully the respective subjects and the conditions bearing on them. It should be borne in mind that the recommendations, which have been founded on data obtained directly from the industries, are necessarily compromises in greater or less degree of the current practice of individual companies, and involve what may be considered ideal practice so far as this is feasible. It should be understood also that the recommendations need not necessarily be put into practice upon their adoption by the Society, but that they should be followed by the various industries when changes in design or production make it economically possible to do so.

The members of the Standards Committee, who are selected because of their broad experience and ability to represent adequately the many branches of the automotive industries, have given generously of their time and resources to the standardization work during the past months in face of their more or less increased regular duties. The standardization work involved traveling to and attending 24 Division meetings and a dozen or more Subdivision meetings.

This truly indicates the recognized value of standards, and makes it practically necessary for the managers and executive directors of the industries to keep informed of the work being done by the Society, and to give it due recognition and support by having the standards used in their products as extensively as possible. The manufacturing and sales executives are urged to review the recommendations of the Standards Committee, to bring their points of view to bear in the discussion at the White Sulphur sessions and to facilitate the work of the engineers in the production of the products involved by having the standard recommendations reduced to practice as promptly and widely as is feasible.

PROGRESS REPORTS

Arrangements have been made with some of the Chairmen of Divisions and Subdivisions for presenting progress reports on a few of the more important subjects that are under consideration but are not ready for final recommendation to the Standards Committee. These reports are intended primarily to familiarize the Standards Committee members in a general way with what is being accomplished and to give the Divisions or Subdivisions the benefit of discussion by the Standards Committee and its guests as a guide in further study of the subjects.

The subjects on which it is planned to have progress discussion are listed below. The progress report of the Lubricants Division is appended to the Division reports.

Crankcase Lubricants
- H. C. Mougey, chairman Lubricants Division

Engine and Frame Numbers
- R. S. Begg, chairman Passenger-Car Division
- J. B. Fisher, chairman Engine Division

Metric Thrust Ball-Bearings
- E. N. Carter, member Subdivision

Brake-Linings
- Clarence Carson, chairman Subdivision

Automotive Starting and Lighting Equipment
- A. D. T. Libby, member Electrical Equipment Division

The presentation of the last-named subject in the foregoing list at the Standards Committee meeting is planned in order to discuss the work that has been done in the standardization of automotive starting and lighting equipment and the actual use of the standards by engine and automobile manufacturers. Mr. Libby is well posted on the work of the Electrical Equipment Division and other bodies interested in the subject, which has resulted in the standards adopted by the Society and printed on pages B17 to B20 inclusive of the S.A.E. HANDBOOK.

AXLE AND WHEELS DIVISION REPORT

Division Personnel

G. W. Dunham, *Chairman*	Savage Arms Corporation
C. C. Carlton, *Vice-Chairman*	Motor Wheel Corporation
R. S. Begg	Jordan Motor Car Co.
T. V. Buckwalter	Timken Roller Bearing Co.
A. C. Burch	Formerly with Clydesdale Motor Truck Co.
R. J. Burrows	Clark Equipment Co.
L. W. Close	Bock Bearing Co.
J. Coapman	Russel Motor Axle Co.
C. S. Dahlquist	Eaton Axle Co.
F. S. Denneen	Grant Motor Car Co.
F. W. Gurney	Gurney Ball Bearing Co.
F. P. Hall, Jr.	Salisbury Axle Co.
G. W. Harper	Columbia Axle Co.
G. L. Lavery	West Steel Casting Co.
A. M. Laycock	Sheldon Axle & Spring Co.
H. V. Ludwick	Budd Wheel Corporation
C. T. Myers	Consulting Engineer
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
H. Vanderbeek	Timken-Detroit Axle Co.

MOTOR-TRUCK FRONT-AXLE HUBS

(Proposed Extension of S.A.E. Recommended Practice)

When the present S.A.E. Recommended Practice for Motor-Truck Front-Axle Hubs, page F1b of the S.A.E. HANDBOOK, was reported to the Standards Committee in January 1921, the report contained many detail dimensions that were not included in the recommended practice finally adopted in July.

In January 1922 a new Division of the Standards Committee, known as the Axle and Wheels Division, was appointed to consider all matters pertaining to axle and wheel assemblies. Among the subjects assigned to this Division is Passenger-Car Front-Axle Hubs, a recommendation for which is now being formulated by a Subdivision, which has practically the same personnel as the Subdivision of the Truck Division that formulated the Report on Motor-Truck Front-Axle Hubs. At a meeting of this Subdivision in April it was recommended that the complete dimensions for the threaded end of the spindle for motor-truck front-axle hubs, that were shown in the original report last year, should be incorporated in the present S.A.E. Recommended Practice to provide a complete standard for motor-truck front-axle spindles. It was also recommended that a supplementary table, to be used as general information only, be appended to the recommended practice to indicate the fundamental factors upon which the report was based. These recommendations were approved by the members of the Axle and Wheels Division by letter ballot. Therefore

> The Axle and Wheels Division recommends that the accompanying tables be approved by the Standards Committee for adoption as an extension of the present S.A.E. Recommended Practice for Motor-Truck Front-Axle Hubs, page F1b of the S.A.E. HANDBOOK

DIMENSIONS OF MOTOR-TRUCK FRONT-AXLE SPINDLE-ENDS

Spindle No.	5	6	7	8	9
Thread, U. S. S.	1⅛–7	1⅛–7	1½–6	1½–6	1½–6
Length of threaded end	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{13}{16}$	$1\frac{13}{16}$	$1\frac{13}{16}$
Height from center-line of spindle to flat milled on threaded-end	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$
Center of cotter-pin hole to end of thread	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$
Diam. of cotter-pin hole	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{7}{32}$
Width of groove at back end of thread	⅛	⅛	⅛	⅛	⅛

GENERAL INFORMATION FOR MOTOR-TRUCK FRONT-AXLE HUBS

Spindle No.	5	6	7	8	9
Spindle load rating in Lb. on solid tire at ground	1250	1625	2125	2750	3500
Solid tire size	34x3½	36x4	36x5	36x6	36x7
Solid tire load rating, Lb.	1300	1700	2500	3300	4200

ELECTRICAL EQUIPMENT DIVISION REPORT

Division Personnel

F. W. Andrew, *Chairman*	Eisemann Magneto Corporation
T. L. Lee, *Vice Chairman*	North East Electric Co.
Azel Ames	Kerite Insulated Wire & Cable Co.
G. S. Cawthorne	Master Trucks, Inc.
W. A. Chryst	Dayton Engineering Laboratories Co.
S. F. Evelyn	Continental Motors Corporation
C. F. Gilchrist	Electric Auto-Lite Corporation
W. S. Haggott	Packard Electric Co.
C. H. Kindl	Westinghouse Electric & Mfg. Co.
F. C. Kroeger	Remy Electric Co.
B. M. Leece	Leece-Neville Co.
A. D. T. Libby	Splitdorf Electrical Co.
Charles Marcus	Bijur Motor Appliance Co.
Ernest Wooler	Cleveland Automobile Co.

IGNITION-DISTRIBUTOR MOUNTINGS

(Proposed Revision of S.A.E. Standard)

The present S.A.E. Standard for Ignition-Distributor Mountings, page B16 of the S.A.E. HANDBOOK, specifies that the distance from the base of the distributor mounting barrel to the bottom of the tongue of the distributor half of the driving coupling shall "vary to suit conditions." As it was suggested by an ignition-distributor manufacturer that this dimension should be definitely established, W. A. Chryst, of the Dayton Engineering Laboratories Co., was appointed a Subdivision to report on the advisability of this.

Mr. Chryst submitted at the April meeting of the Electrical Equipment Division a report which was approved.

This report was referred to automobile engine and ignition-apparatus manufacturers for comment and met with general approval. Therefore

> The Division recommends that the present S.A.E. Standard for Ignition-Distributor Mountings be revised by
>
> (1) Specifying a dimension of 27/32 in. for the distance from the bottom of the distributor mounting barrel to the bottom of the tongue on the distributor half of the driving coupling of the Type-B ignition-distributor
>
> (2) Changing the limits for the bore in the coupling from 0.4930 in. maximum and 0.4920 in. minimum to 0.4915 in. maximum and 0.4905 in. minimum

MAGNETO MOUNTINGS

(Proposed S.A.E. Standard)

Several requests were received by the Society in 1920 for the standardization of mountings for magnetos used on stationary engines, isolated electric-lighting plants and small tractors on which the present S.A.E. Standard magneto mounting could not be used. In view of the importance of such standardization, the following Subdivision of the Electrical Equipment Division was appointed:

F. W. Andrew, *Chairman*	Eisemann Magneto Corporation
W. F. Borgerd	International Harvester Co.
James M. Edwards	Associated Manufacturers Co.
A. D. T. Libby	Splitdorf Electrical Co.
F. L. Tubbs	Alamo Farm Light Co.

As the three members last named are members of the Agricultural Power Equipment, Stationary-Engine and the Isolated Electric Lighting Plant Divisions, respectively, the Subdivision was representative of all interests involved.

Upon making a general survey of the situation, the Subdivision found that the practice varied considerably,

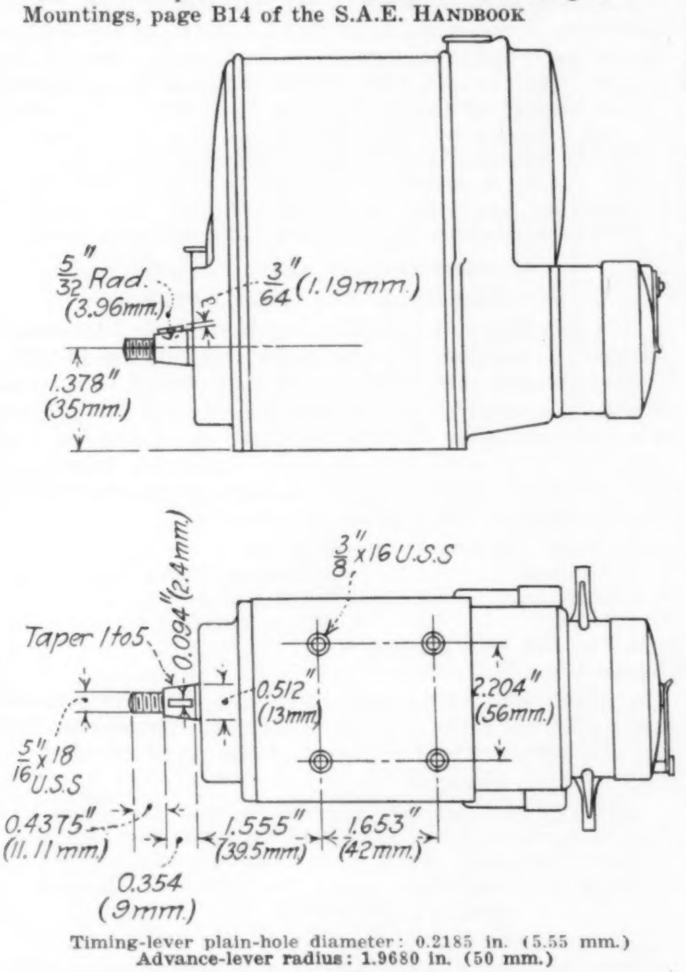
comparatively few companies using the same dimensions for mounting small magnetos.

A meeting of members of the Subdivision and a number of others was held on Nov. 3, 1921, following a two-day session of the Gas Engine and Farm Power Association at which most of the magneto manufacturers were represented. As a result of this conference a Subdivision report was formulated and sent to stationary-engine, isolated electric-lighting plant, tractor and magneto manufacturers for comment. A large number of replies were received indicating general approval of the report.

The recommendation was also published in the April issue of THE JOURNAL.

At the meeting of the Electrical Equipment Division on April 14, Mr. Andrews submitted the report of the Subdivision which was carefully discussed. It was thought, however, advisable to extend the report to include the timing-lever dimensions in accordance with the dimensions specified in the present standard for motorcycle magnetos. Therefore

> The Division recommends that the magneto dimensions given in the accompanying illustration be adopted as S.A.E. Recommended Practice, as an addition to the present S.A.E. Standard for Magneto Mountings, page B14 of the S.A.E. HANDBOOK



Timing-lever plain-hole diameter: 0.2185 in. (5.55 mm.)
Advance-lever radius: 1.9680 in. (50 mm.)

BREAKER-CONTACTS

(Proposed S.A.E. Recommended Practice)

At the October 1921 meeting of the Electrical Equipment Division it was suggested by a Division member that a standard for breaker-contact nuts would overcome the necessity for service-stations carrying several sizes of wrenches in stock to fit the different makes of breaker-contact points.

As the importance of attaining such standardization in practice was appreciated, information was obtained from ignition-apparatus manufacturers as to their current practice. A careful analysis of the information obtained was made at a meeting of the Division in April which indicated that only the distance across the flats of the hexagon-head and the size and pitch of the thread should be standardized, as it was not intended to provide in the standard for complete interchangeability of breaker-contacts of different makes. Therefore

> The Electrical Equipment Division recommends for S.A.E. Recommended Practice that the hexagon-head of breaker-contacts and check-nuts shall be ¼ in. across flats and that the threads shall be No. 10-40 or No. 8-40

STARTING-MOTOR FLANGE MOUNTINGS

(Proposed Revision of S.A.E. Standard)

The Electrical Equipment Division recommended at the Standards Committee meeting in January 1922 that the bolt-hole diameters for the Nos. 1 and 2 mountings specified in the S.A.E. Standard for Starting-Motor Flange Mountings, page B19 of the S.A.E. HANDBOOK, be changed from 13/32 to 7/16 in. This revision was approved and adopted by the Society in March.

Several negative letter-ballots were cast by Society Members against this change. The reasons given in support of the negative votes were referred to and carefully considered by the Subdivision on Generator and Starting-Motor Mountings.

At the meeting of the Electrical Equipment Division in April the Subdivision reported that, after an investigation of the reasons for making the change in the standard last March, it felt that an error had been made in adopting the 7/16-in. hole diameter. Therefore

> The Electrical Equipment Division recommends that dimension "C," the diameter of the bolt-holes for sizes Nos. 1 and 2 of the S.A.E. Standard for Starting-Motor Flange Mountings, page B19 of the S.A.E. HANDBOOK, be changed from 7/16 to 13/32 in.

AGRICULTURAL POWER EQUIPMENT DIVISION

Division Personnel

John Mainland, *Chairman*	Advance-Rumely Co.
C. B. Rose, *Vice-Chairman*	Moline Plow Co.
D. L. Arnold	International Harvester Co.
J. B. Bartholomew	Avery Co.
J. H. Davis	General Motors Research Corporation
C. E. Frudden	Hart-Parr Co.
A. H. Gilbert	Rock Island Plow Co.
R. O. Hendrickson	J. I. Case Plow Works Co.
M. B. Morgan	Cleveland Tractor Co.
A. W. Scarratt	Minneapolis Steel & Machinery Co.
O. W. Sjogren	University of Nebraska
William Turnbull	Holt Mfg. Co.
L. W. Witry	Waterloo Gasoline Engine Co.
G. A. Young	Purdue University
O. W. Young	Hyatt Roller Bearing Co.

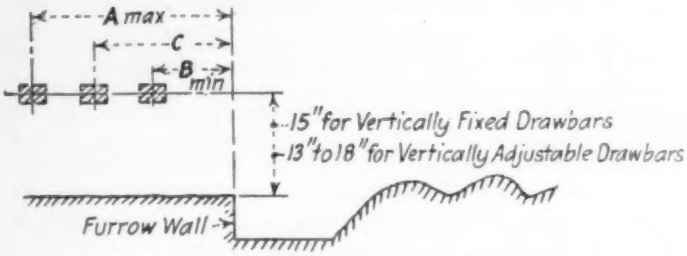
TRACTOR DRAWBAR ADJUSTMENTS

(Proposed Revision of S.A.E. Standard)

Since the adoption of the last revision of the S.A.E. Standard for Tractor Drawbar Adjustments, page K40 of the S.A.E. HANDBOOK, O. B. Zimmerman of the International Harvester Co., who has been serving as a Subdivision, carried out further tests with the result that he recommended some slight changes in the present standard at a meeting of the Agricultural Power Equipment Division in April. Other members of the Division reported that, although the results of similar tests carried

out by them did not exactly agree with Mr. Zimmerman's, they were within the limits given in the proposed revision. Therefore

> The Agricultural Power Equipment Division recommends that the present S.A.E. Standard for Tractor Drawbar Adjustments, page K40 of the S.A.E. HANDBOOK, be revised to conform with the limits specified in the accompanying table



TRACTOR DRAWBAR ADJUSTMENTS

PLOW	TWO-BOTTOM		THREE-BOTTOM		FOUR-BOTTOM	
	12-In.	14-In.	12-In.	14-In.	12-In.	14-In.
A Max.	30	30	30	32	35	40
B Min.	13	15	18	20	22	24
C Best Average	17	20	22	26	27	32

A—Maximum hitching position from the furrow wall, very often necessary when the tractor operates on the land

B—Minimum hitching position from the furrow wall when the tractor operates in the furrow

C—Best average hitching position from the furrow wall

ENGINE DIVISION REPORT

Division Personnel

J. B. Fisher, *Chairman*	Waukesha Motor Co.
R. J. Broege, *Vice-Chairman*	Buda Co.
P. J. Dasey	Midwest Engine Co.
S. F. Evelyn	Continental Motors Corporation
E. J. Hall	Hall-Scott Motor Co.
H. B. Massey	Holmes Automobile Co.
A. F. Milbrath	Wisconsin Motor Mfg. Co.
Louis Schwitzer	Automotive Parts Co.
M. J. Steele	Packard Motor Car Co.
William Turnbull	Holt Mfg. Co.

CRANKCASE DRAIN-PLUGS

(Proposed S.A.E. Recommended Practice)

The advisability of designing engines with a conveniently located oil-pan drain that would permit draining the oil from the crankcase as easily as it is poured into it was brought to the attention of the Society in the fall of 1921. The subject was referred to the Engine Division for consideration.

Information was obtained from engine and automobile builders who build their own engines as to their methods of draining crankcases. Suggestions were submitted by an oil company that had conducted an investigation of the subject which resulted in a large number of suggestions from engineers.

Data obtained as to the size of the oil-drain hole and the location and means of operating the drain-plug were analyzed carefully at a meeting of the Engine Division in April. It was not considered feasible to standardize upon a definite means of operating the drain-plug or the design of the oil drain. The following recommendation was considered, however, to be in accord with good engineering practice. Therefore

> The Engine Division recommends for S.A.E. Recommended Practice that crankcase drain-plugs shall have a minimum opening of ¾ in., be located at the lowest point of the oil-pan and be operable from under the engine hood

FLYWHEEL HOUSINGS

(Proposed Revision of S.A.E. Standard)

The present S.A.E. Standard for Flywheel Housings, page A1 of the S.A.E. HANDBOOK, was revised in July 1921 to specify that

> The minimum diameter of the clearance space for crankshaft flywheel bolts shall be 6⅛ in. and the minimum depth ⅝ in.

At the September 1921 meeting of the Division this revision was discussed and the point brought out that the clearance space allowed would not be satisfactory for two-bearing engines having a large crankshaft flange. It was thought that the present clearances would not permit the use of ½-in. U. S. Standard bolts and nuts or S.A.E. Standard bolts and nuts with cotter-pins. Furthermore, it was considered that a larger clearance space should be recommended to allow for possible wear in the clutch. The revision indicated was favorably considered at the meeting of the Division in April 1922. Therefore

> The Engine Division recommends that the present S.A.E. Standard for Flywheel Housings, page A1 of the S.A.E. HANDBOOK, be revised to specify that the clearance space for crankshaft flywheel bolts shall be 6½-in. minimum diameter and ¾-in. minimum depth

MOTORCYCLE CARBURETER FLANGES

(Proposed S.A.E. Recommended Practice)

The possibility of standardizing motorcycle carbureter flanges was suggested by the Motorcycle Division in 1920. It was thought that, although practice varied to a large extent, a standard could be established, and the subject was referred to the Engine Division. A subdivision was appointed that conducted a rather extensive investigation among the carbureter and motorcycle manufacturers. The members of the Subdivision were:

R. J. Broege, *Chairman*	Buda Co.
E. H. Shepard	Stromberg Motor Devices Co.
V. I. Shobe	Zenith Carbureter Co.
W. G. Clark	Wilcox-Bennett Carbureter Co.
C. B. Franklin	
Hendee Mfg. Co.	

A report was formulated which met with the general approval of the carbureter and motorcycle manufacturers, and was reviewed and approved at the April meeting of the Division. Therefore

> The Engine Division recommends that the dimensions specified in the accompanying tables for Motorcycle Carbureter Flanges be added to the present S.A.E. Recommended Practice for Carbureter Flanges, page A8 of the S.A.E. HANDBOOK

The proposed two-bolt type flanges for motorcycle carbureters conform with the present standard two-bolt type flanges. The three-bolt flanges have been proportioned so that all nominal carbureter sizes of this type can be made from but two castings.

IRON AND STEEL DIVISION REPORT

Division Personnel

F. P. Gilligan, *Chairman*	Henry Souther Engineering Corporation
W. C. Peterson, *Vice-Chairman*	Electric Alloy Steel Co.

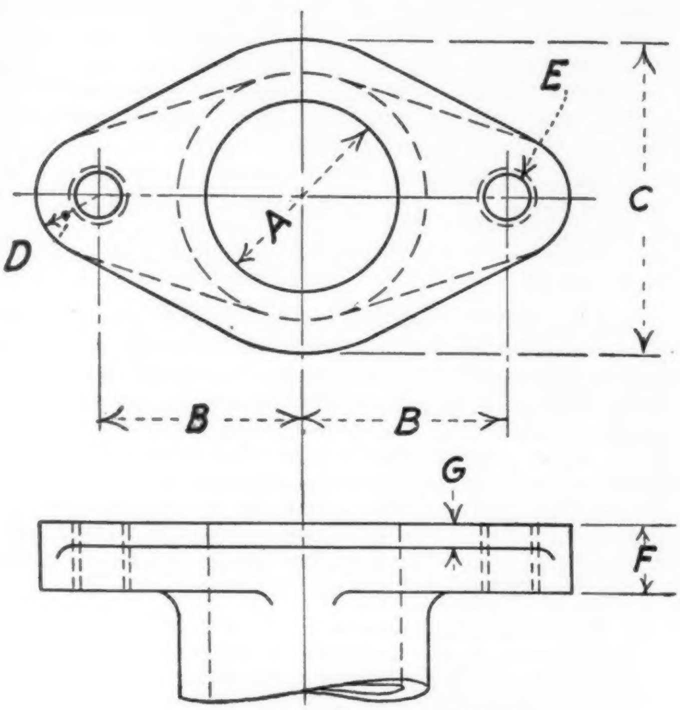


FIG. 1—3/4 AND 7/8-IN. NOMINAL SIZE

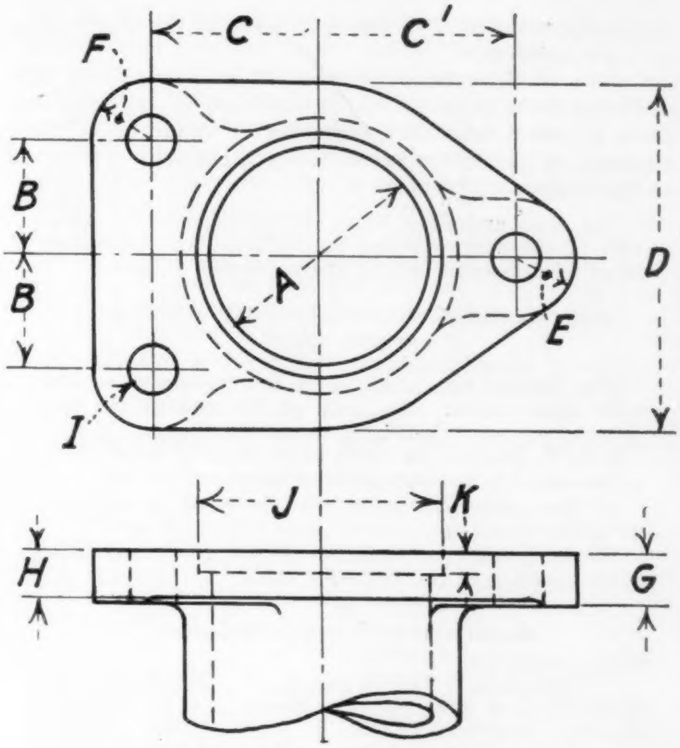


FIG. 2—1-IN. NOMINAL SIZE

Nominal Carbureter Size	FIG. No. 1 A	B	C	D	E	F	G	FIG. No. 2 A	B	C	C[1]	D	E	F	G	H	I	J	K
3/4	1 1/16	1 1/8	1 3/4	11/32	5/16–18	3/8	1/8	1 1/16	5/8	29/32	1 1/16	1 7/8	5/16	5/16	17/64	1/4	0.257 Drill	1 11/32	0.100
7/8	1 1/16	1 1/8	1 3/4	11/32	5/16–18	3/8	1/8	1 1/16	5/8	29/32	1 1/16	1 7/8	5/16	5/16	17/64	1/4	0.257 Drill	1 11/32	0.100
1	1 3/16	1 3/16	1 7/8	11/32	5/16–18	13/32	5/32	1 3/16	5/8	29/32	1 1/16	1 7/8	5/16	5/16	17/64	1/4	0.257 Drill	1 11/32	0.100
1 1/4								1 7/16	3/4	1 1/32	1 1/4	2 1/8	5/16	5/16	1/4		5/16–18	1.60	0.100

R. M. Bird	Bethlehem Steel Co.
H. T. Chandler	C. H. Wills & Co.
A. L. Colby	Consulting Metallurgist
L. A. Danse	Cadillac Motor Car Co.
C. N. Dawe	Studebaker Corporation of America
B. H. DeLong	Carpenter Steel Co.
A. P. Eves	International Harvester Co.
E. L. French	Crucible Steel Co. of America
H. L. Greene	Willys-Overland Co.
C. G. Heilman	General Motors Corporation
E. J. Janitzky	Illinois Steel Co.
J. B. Johnson	Air Service
F. C. Langenberg	Ordnance Department
A. H. Miller	Midvale Steel & Ordnance Co.
C. S. Moody	Minneapolis Steel & Machinery Co.
J. H. Nelson	Wyman-Gordon Co.
G. L. Norris	Vanadium Corporation of America
W. H. Phillips	R. D. Nuttall Co.
S. P. Rockwell	Consulting Metallurgist
M. P. Rumney	Detroit Steel Products Co.
C. F. W. Rys	Carengie Steel Co.
R. B. Schenck	Buick Motor Co.
M. H. Schmid	United Alloy Steel Corporation
H. J. Stagg	Halcomb Steel Co.
J. M. Watson	Hupp Motor Car Corporation
J. H. G. Williams	Billings & Spencer

LEAF-SPRING STEEL

(Proposed S.A.E. Recommended Practice)

In 1921 the leaf-spring manufacturers, acting through the Motor & Accessory Manufacturers Association, submitted for approval by the Society a proposed specification for rolling tolerances for automobile leaf-spring steel. The specification had been prepared and approved by steel and leaf-spring manufacturers in view of the need for uniformity in rolling leaf-springs.

The subject was assigned to the Iron and Steel Division and considered at a meeting in January 1922. The proposed specification was then circulated, with favorable results, among steel mills, automobile and spring manufacturers, and also referred to the Springs Division,

which approved it. It was published in the April issue of THE JOURNAL.

Copies of all comments received as a result of this publicity were referred to the members of the Iron and Steel Division, who have indicated by letter ballot their approval of the recommendation as originally submitted to the Society. Therefore

The Iron and Steel Division recommends the adoption of the accompanying specification for leaf-spring steel tolerances as S.A.E. Recommended Practice

ROLLING TOLERANCES FOR CONCAVE AUTOMOBILE SPRING STEEL

The finished bars shall be of double-concave section with round edges. The radii of the arcs of the two concave surfaces shall be of equal length

Rolls to produce the round edges shall be turned to a radius equal to two-thirds the thickness of the bar

All bars ordered to gage shall be rolled to the Birmingham wire gage

All bars must meet the width and thickness tolerances specified in the following table

WIDTH AND THICKNESS TOLERANCES

Width of Flat, in. Over	Width of Flat, in. To, Inclusive	Width, in. Plus	Width, in. Minus	Thickness,[1] in. Plus	Thickness,[1] in. Minus
0	2¼	1/32	0	0.005	0.005
2¼	3	3/64	0	0.006	0.006
3	5	1/16	0	0.007	0.007

[1] Thickness measurement to be taken at edge of bar where concave surface intersects round edge.

The difference in thickness between the two edges of each bar shall not be greater than those given in the following table

DIFFERENCES IN THICKNESS

Width of Flat, in. Over	Width of Flat, in. To, Inclusive	Difference in Thickness, in.
0	2	0.002
2	3	0.003
3	5	0.004

Spring-steel bars shall not have more than 1-in. curvature in 20 ft., or 1¼ in. in 25 ft., or 1½ in. in 30 ft.

The concavity, or difference between the thickness at the edges and at the center of the bar, shall be as specified in the following table

ALLOWABLE VARIATIONS IN CONCAVITY

Width, in.	Nominal Concavity, in.	Maximum Concavity, in.	Minimum Concavity, in.
1½	0.007	0.009	0.004
1¾	0.008	0.010	0.005
2	0.010	0.012	0.006
2¼	0.011	0.013	0.007
2½	0.013	0.015	0.009
3	0.016	0.018	0.012
3½	0.018	0.020	0.013
4	0.021	0.023	0.016
5	0.029	0.031	0.023

STEEL SPRING-WIRE

(Proposed S.A.E. Standard)

At the time of the Society letter-ballot on the adoption of standards approved at the Standards Committee Meeting in January 1922, an objection was made that the chemical specification for carbon steel did not provide proper compositions for drawn spring-wire. This objection was considered at a meeting of the Division in February 1922. Although the specifications formulated by the Iron and Steel Division so far have included primarily steel compositions for forging purposes only, it was felt that chemical compositions might well be adopted for spring steel-wire. A representative Subdivision was appointed, the members of which were:

L. A. Danse, *Chairman*	Cadillac Motor Car Co.
W. C. Peterson	Atlas Crucible Steel Co.
E. W. Stewart	William B. Gibson Co.
F. C. Elder	American Steel & Wire Co.
H. S. Durant	American Steel & Wire Co.

The Subdivision held a meeting on April 10 at which a recommendation was approved, and this was submitted to and approved by the Division. Copies of the recommendation were sent to automobile and spring manufacturers for comment, the result being favorable in general. Some suggestions were received to the effect that suitable specifications should be adopted for poppet-valve spring-wire made of other than carbon steels, such as chrome-vanadium, which would withstand fatigue in high-duty service. These suggestions have been referred to the Iron and Steel Division. Therefore

The Division recommends that the chemical compositions given in the accompanying table be adopted as S.A.E. Standard for round, cold-drawn wire up to 3/16-in. diameter, except for some types of springs used in clutches, which are hot-rolled

STEEL WIRE FOR HELICAL SPRINGS

S.A.E. Steel No.	Wire Diameters, In.[2]	Carbon	Manganese	Phosphorus Max.	Sulphur Max.	Silicon Max.
1350	0.025 to 0.100	0.45–0.55	0.90–1.20	0.04	0.05	0.30
1360	0.100 and larger	0.55–0.70	0.90–1.20	0.04	0.05	0.30

[2] Approximately.

LIGHTING DIVISION REPORT

Division Personnel

W. A. McKay, *Chairman*	Westinghouse Lamp Co.
C. A. Michel, *Vice-Chairman*	Guide Motor Lamp Mfg. Co.
J. T. Caldwell	National Lamp Works
C. E. Godley	Edmunds & Jones Corporation
C. A. B. Halvorson, Jr.	General Electric Co.
L. C. Porter	Edison Lamp Works
E. S. Preston	Chicago Electric Mfg. Co.
C. D. Ryder	Corcoran-Victor Co.
C. H. Sharp	Electrical Testing Laboratories
J. C. Stearns	Culver-Stearns Mfg. Co.
T. I. Walker	Providence Base Works
E. E. Wood	Miniature Incandescent Lamp Corporation
Ernest Wooler	Cleveland Automobile Co.

HEAD-LAMPS

(Proposed Revision of S.A.E. Recommended Practice)

The present S.A.E. Recommended Practice on page B1 of the S.A.E. HANDBOOK is not considered to be entirely clear in the paragraph under the sub-heading "Mounting." It is also felt by the Division that specific mention should be made in this specification of adjustable types of mounting bracket on radiator shells, tie-rods and the like. Therefore

The Lighting Division recommends that the present S.A.E. Recommended Practice for Head-lamps, page B1 of the S.A.E. HANDBOOK, be revised as indicated below

(1) Change the second sentence in the paragraph under the subheading "Mounting," which

now reads. "Means for adjusting the head-lamps shall be provided so as to permit a change in the vertical as well as in the horizontal angle of the head-lamps, * * *" to read "For adjustable types of head-lamp mountings means shall be provided, * * *"

(2) Under the sub-heading "Brackets" insert the following: "*Adjustable Type.*—The adjustable type of mounting-bracket for headlamps mounted on fenders, radiator shells, frame-pillars, tie-rods and similar constructions, as shown on page B1a of the S.A.E. HANDBOOK, is recommended"

ELECTRIC INCANDESCENT LAMPS

(Proposed Revision of S.A.E. Standard)

At the meeting of the Lighting Division held on May 3 information was received indicating that the G10, G12 and G16½ sizes of focusing type electric incandescent lamp have practically been discontinued in use. It was the opinion that the present S.A.E. Standard on page B3 of the S.A.E. HANDBOOK should be revised by cancelling these lamp sizes. Therefore

The Lighting Division recommends that the G10, G12 and G16½ lamps be omitted from the S.A.E. Standard for Electric Incandescent Lamps, page B3 of the S.A.E. HANDBOOK

ELECTRIC INCANDESCENT LAMP VOLTAGE

(Proposed Revision of S.A.E. Standard)

In connection with the present S.A.E. Standard on page B1c of the S.A.E. HANDBOOK, which specifies four voltage-ranges, namely 6 to 8, 8 to 10, 12 to 16, and 18 to 24, for automotive starting and lighting equipment, it is understood that 8 to 10 and 18 to 24-volt lighting equipment has been discontinued by the manufacturers and that all substantially standard equipment is now being made for 6 to 8 and 12 to 16 voltage-ranges. Therefore

The Lighting Division recommends that the present S.A.E. Standard for Electric Incandescent Lamp Voltages be revised by omitting the 8 to 10 and the 18 to 24 voltage-ranges and the reference to four and nine battery-cell arrangements

MOTORBOAT LIGHTING VOLTAGES

(Proposed S.A.E. Recommended Practice)

At the meeting of the Motorboat Division held in New York City on April 10 it was suggested that standard voltages be adopted for lighting equipment on motorboats, inasmuch as there is no well-defined practice in this connection.

The discussion indicated that two systems are used for lighting motorboats; (*a*) combined starting and lighting apparatus and (*b*) separate lighting units. It was felt that 12 volts would be most suitable for the combined systems, to make standard automobile types of starting and lighting apparatus available, the 6-volt system being too low in voltage for the usual comparatively long lighting circuits.

It was pointed out that the usual practice for lighting the larger motorboats is to install separate lighting equipment, and that the best voltages are already standard for isolated electric-lighting plants and the many electric lamps, fittings and fixtures operated with them.

The subject was referred to the Motorboat Lighting Equipment Subdivision of the Lighting Division, which reported to the Lighting Division on May 3. The report covered a nominal 6-volt combined starting and lighting system for use in the smaller types of motorboat, where the length of the circuits is relatively short. The 64-volt system was excluded as its use is decreasing, especially in railroad-car lighting systems where it is used most generally. Sixty-four-volt lamps and other types of fixture also are being discontinued. Therefore

The Lighting Division recommends that the accompanying report submitted by the Subdivision be adopted as S.A.E. Recommended Practice.

(1) For motorboats and small cruisers having combined starting and lighting equipment, it is recommended that nominal 6-volt (6 to 8 volts) or nominal 12-volt (12 to 16 volts) systems be used

(2) For larger cruisers having separate lighting equipment, it is recommended that the 32 or 110-volt system be used

NON-FERROUS METALS DIVISION REPORT

Division Personnel

Charles Pack, *Chairman*	Doehler Die-Casting Co.
W. B. Price, *Vice-Chairman*	Scovill Mfg. Co.
W. H. Bassett	American Brass Co.
A. G. Carman	Franklin Die-Casting Corporation
D. L. Colwell	Stewart Mfg. Co.
G. K. Elliott	Lunkenheimer Co.
J. R. Freeman, Jr.	Bureau of Standards
E. S. Fretz	Light Mfg. & Foundry Co.
A. J. Hall	General Motors Corporation
Zay Jeffries	Aluminum Manufacturers, Inc.
H. C. Mougey	General Motors Research Corporation
G. C. Rauhauser	Denby Motor Truck Co.
J. F. Thompson	International Nickel Co.
Samuel Tour	Doehler Die-Casting Co.
W. R. Webster	Bridgeport Brass Co.

ALUMINUM ALLOYS

(Proposed S.A.E. Standard)

In 1921 the Society received a request that a specification be formulated for an aluminum alloy with a copper-content of about 10 per cent, commonly used for pistons. This request was referred to Zay Jeffries, of the Aluminum Manufactures, Inc., who is chairman of the Subdivision on Aluminum Alloys. The members of the Subdivision are

Zay Jeffries, *Chairman*	Aluminum Manufactures, Inc.
E. Blough	Aluminum Manufactures, Inc.
E. S. Fretz	Light Mfg. & Foundry Co.

At the May 1922 meeting of the Division, the Subdivision submitted a specification which was approved by the Division. Therefore

The Non-Ferrous Metals Division recommends that Specification No. 34 be adopted as S.A.E. Standard

SPECIFICATION NO. 34, ALUMINUM ALLOYS

Composition in percentage:

Aluminum, min.	87.00
Copper	9.25 to 10.75
Iron	0.90 to 1.50
Magnesium	0.15 to 0.35
All other elements, not over	0.75

General Information.—Test-bars cast in a chill mold show a tensile-strength of 24,000 to 30,000 lb. per sq. in. and the elongation in 2 in. is usually less than 1 per cent. The specific gravity should not exceed 2.95. The Brinell hardness number, using a 500 or 1000-kg. load with a ball 10 mm. in diameter, should be not less than 85 and should average about 105.

This alloy cast in permanent molds is used principally for pistons, camshaft bearings, valve tappet guides and other parts where high hardness and good bearing qualities are essential.

WROUGHT NON-FERROUS ALLOYS

(Proposed S.A.E. Standard)

Since the last Non-Ferrous Metals Division report was approved by the Society, the Subdivision on Wrought Non-Ferrous Alloys has formulated additional specifications for wrought alloys used extensively in the automotive industry. The members of this Subdivision are

W. B. Price, *Chairman*	Scovill Mfg. Co.
H. C. Mougey	General Motors Research Corporation
W. H. Bassett	American Brass Co.
William R. Webster	Bridgeport Brass Co.

Three specifications were prepared and submitted by the Subdivision, (*a*) Phosphor-Bronze Strips for Flat Springs, (*b*) Brass Wire for Brazing and (*c*) Soft or Annealed Copper Wire. The Non-Ferrous Metals Division at its meeting on May 1 approved these specifications.

At the meeting of the Division on May 1 the Subdivision on Aluminum Alloys submitted a specification for aluminum sheet, closely conforming to Specification No. B-25-19T, of the American Society for Testing Materials, for adoption as S.A.E. Specification No. 78. This specification was approved by the Division. Therefore

> The Non-Ferrous Metals Division recommends that Specifications Nos. 77, 78, 82 and 83 be adopted as S.A.E. Standard

SPECIFICATION No. 77, PHOSPHOR BRONZE STRIP FOR FLAT SPRINGS

This specification covers bronze strips up to 0.080 in. gage for flat springs and includes a variety of tempers in two alloys, from which a suitable quality can be chosen for any ordinary purpose.

Composition in percentage:

	Grade A	Grade B
Tin	4.00 to 6.00	7.00 to 9.00
Phosphorus	0.03 to 0.40	0.03 to 0.20
Zinc, max.	0.20	0.20
Iron, max.	0.10	0.10
Lead, max.	0.10	0.10
Copper	Remainder	Remainder

Temper Designation.—The temper of strip phosphor bronze shall be designated as follows:

Temper	Reduction, B. & S., Nos.
Half Hard	2
Hard	4
Extra Hard	6
Spring	8

Physical Properties.—The average tension test of two samples of sheet thinner than 0.080 in. should conform to the minimum requirements specified in the accompanying table. In very thin strips, on account of difficulties in testing, the elongation may be considerably less than the values given.

MINIMUM PHYSICAL PROPERTIES

Temper	GRADE A		GRADE B	
	Minimum Tensile-Strength, Lb. Per Sq. In.	Minimum Elongation in 2 In., Per Cent	Minimum Tensile-Strength, Lb. Per Sq. In.	Minimum Elongation in 2 In., Per Cent
Half Hard	55,000	15.0	65,000	20.0
Hard	75,000	5.0	85,000	7.5
Extra Hard	85,000	2.0	100,000	1.0
Spring	90,000	1.0	......	....

Dimensional Tolerances.—The width of the strip shall not vary more than 0.01 in. from the size specified in the order.

General Information.—These should be considered as general specifications. Phosphor bronze strip is used for various kinds of springs where the manufacturing requirements and the uses to which the springs are put are too particular to be specified by ordinary physical tests. It is advisable to submit samples or drawings to the manufacture and secure an adjustment of temper to suit the manufacturing and service requirements of the article.

Flat springs formed with easy bends across the grain are usually made of the Grade A alloy, "Spring" temper.

Flat springs with easy bends either across or with the grain are usually made of the Grade B alloy, "Extra Hard" temper.

Clips or contact springs with most difficult bends are usually made of the Grade B alloy, "Hard" temper.

SPECIFICATION No. 78, ALUMINUM SHEET AND STRIP

Composition in percentage:

Aluminum, min. 99.00

Physical Properties.—Aluminum sheet and strip are furnished in several tempers or degrees of hardness. The mechanical properties of aluminum sheet or strip conforming to Tempers, No. 1, Soft Annealed; No. 2, Half-Hard, and No. 3, Hard, are given in Table 1.

The tension test-specimen is taken parallel to the direction of cold-rolling. Sheet or strip of Temper No. 1 should withstand being bent double in any direction and hammered flat. Sheet or strip of Temper No. 2 should withstand being bent around a pin of radius equal to the thickness of the sheet,

THICKNESS TOLERANCES

Thickness				Width, In.			
B. & S. Gage No.		Decimal		Up to 5 Incl.	Over 5 to 8 Incl.	Over 8 to 11 Incl.	Over 11 to 14 Incl.
From	To, Incl.	From	To, Incl.				
0000	0	0.4600	0.3248	0.0044	0.0048	0.0051	0.0055
0	4	0.3248	0.2043	0.0039	0.0043	0.0046	0.0050
4	8	0.2043	0.1284	0.0034	0.0038	0.0041	0.0045
8	14	0.1284	0.0640	0.0029	0.0033	0.0036	0.0040
14	18	0.0640	0.0403	0.0025	0.0029	0.0033	0.0037
18	24	0.0403	0.0201	0.0020	0.0024	0.0028	0.0032
24	28	0.0201	0.0126	0.0016	0.0020	0.0024	0.0028
28	32	0.0126	0.0079	0.0013	0.0017	0.0020	0.0024
32	35	0.0079	0.0056	0.0010	0.0014	0.0017	0.0022
35	38	0.0056	0.0039	0.0008	0.0012	0.0015	0.0019

All tolerances plus or minus.

TABLE 1—MECHANICAL PROPERTIES FOR TEMPERS NO. 1 TO 3

Temper	B. & S. Gage No.		Thickness, In.		Minimum Tensile-Strength, Lb. Per Sq. In.	Minimum Elongation in 2 In., Per Cent
	From	To, Incl.	From	To, Incl.		
1	12	16	0.0808	0.0509	12,500	30
	17	22	0.0508	0.0227	12,500	20
	23	26	0.0226	0.0159	12,500	10
2	12	16	0.0808	0.0509	18,000	7
	17	22	0.0508	0.0227	18,000	5
	23	26	0.0226	0.0159	18,000	5
3	12	16	0.0808	0.0509	22,000	4
	17	22	0.0508	0.0227	25,000	2
	23	26	0.0226	0.0159	30,000	2

without cracking. Sheet or strip of Temper No. 3 will not endure any considerable bending without cracking.

The usual variations in the thickness of sheet or strip are shown in Table 2.

General Information.—The specific gravity is about 2.70. Young's modulus of elasticity is about 10,000,000 lb. per sq. in.

Aluminum sheet and strip are used for many purposes where the requirements, either because of the service or the forming operations, are too specific to be covered in any general specification. It is usually advisable, therefore, to submit samples or drawings to the manufacturer to assist in the selection of the proper anneal or temper to suit the service or forming operations.

Aluminum sheet or strip is used for automobile bodies, hoods (special flat sheet), fenders, housing covers, floor covering, molding, instrument parts, instrument boards, hubcaps, wire wheel discs, brake-drum covers, miscellaneous pressed parts and for many parts in aircraft construction.

TABLE 2—PERMISSIBLE VARIATIONS

B. & S. Gage No.		Thickness, In.		Permissible Variations, In
From	To, Incl.	From	To, Incl.	
10	17	0.1019	0.0404	0.003
18	26	0.0403	0.0159	0.002

Specification No. 82, Brass Wire for Brazing

Composition in percentage:

Copper	59.00 to 62.00
Lead, max.	0.30
Iron, max.	0.06
Aluminum	None
Zinc	Remainder

This wire should be finished soft annealed. The surface should be clean and free from scale or other foreign materials.

General Information.—Wire in accordance with this specification is suitable for torch welding.

Specification No. 83, Soft or Annealed Copper Wire

Composition.—The copper shall be of such quality and purity that, when drawn and annealed, it shall have the properties and characteristics herein required.

Shapes.—These specifications cover untinned drawn and annealed round wire.

Finish.—The wire must be free from all imperfections not consistent with the best commercial practice.

Necessary brazes in soft or annealed wire must be made in accordance with the best commercial practice.

Specific Gravity.—For the purpose of calculating weights, cross-sections, and other purposes, the specific gravity of copper shall be taken as 8.89 at 20 deg. cent. (68 deg. fahr.).

Dimensions and Permissible Variations.—The size shall be expressed as the diameter of the wire in decimal fractions of an inch.

Wire shall conform to the following permissible variations in nominal diameter:

MECHANICAL REQUIREMENTS FOR SOFT COPPER WIRE

Diameter or Thickness, In.		Tensile-Strength, Lb. Per Sq. In.	Elongation in 10 In., Per Cent
Over	To, Incl.		
0.460	0.290	36,000	35
0.289	0.103	37,000	30
0.102	0.021	38,500	25
0.020	0.003	40,000	20

Wire 0.010 in. diameter and larger, 1 per cent over or under

Wire less than 0.010 in. diameter, 0.1 mil (0.0001 in.) over or under

Mechanical Requirements.—Wire shall be so drawn and annealed that its tensile strength and elongation shall not be greater nor less respectively than the values specified in the accompanying table. For wire whose nominal diameter is between the given sizes, the requirements shall be those of the next larger size included in the table.

Copper wire is usually purchased in B. & S. Gage.

Electrical Resistivity.—Electrical resistivity shall be determined upon fair samples by resistance measurements at a temperature of 20 deg. cent. (68 deg. fahr.), and it shall not exceed 891.58 lb. per mile-ohm.

General Information.—As soft or annealed copper wire is so soft and ductile that it is easily marred by careless handling in the operation of winding or cabling, all testing and inspection must be done at the manufacturer's plant.

While international agreement upon the value 0.15328 ohms per meter-gram at 20 deg. cent. (68 deg. fahr.) for the resistivity of copper equal to 100 per cent conductivity was reached by the International Electro-Technical Commission in 1913, it has been deemed preferable to express the requirements in standard specifications in the terms of quantities directly measurable, rather than by reference to some quantity whose standard value is the subject of agreement only. The use of the arbitrary term "conductivity" has no more warrant than the employment of arbitrary gage numbers. Therefore, in these specifications the requirements are stated as the maximum rejection limits to the resistivity.

For the convenience of those who are accustomed to express resistivity in any one of the several more or less common units, the following equivalents have been prepared for the resistivity of copper at 20 deg. cent. (68 deg. fahr). 891.58 lb. per mile-ohm is equal to 0.15614 ohms per meter-gram, 1.7564 microhms per centimeter-cube, 0.69150 microhms per inch-cube or 10.565 ohms per mil-foot.

White Bearing Metals

(Proposed Extension of S.A.E. Standard)

At the meeting of the Non-Ferrous Metals Division on May 1 it was decided that the limits for the chemical compositions for the babbitt specifications, page D103 of the S.A.E. Handbook, should be closer for babbitt purchased in ingot form than for babbitt in castings; allowance being made for variations in composition due to casting. The present standard specifications are considered satisfactory for casting analyses. Therefore

The Division recommends the adoption of the extended specifications for White Bearing Metals Nos. 11, 12, 13 and 14

SPECIFICATION No. 10, BABBITT

When finished bronze-backed bearings are purchased a maximum of 0.6 per cent lead is permissible in scraped samples provided a lead-tin solder has been used in bonding the bronze and the babbitt.

General Information.—This babbitt is very fluid and may be used for bronze-backed bearings, particularly for thin linings such as are used in aircraft engines. It is also suitable for die castings.

Composition in percentage:

	No. 10 Cast Products	No. 10A Ingots
Tin	90 to 92	90.75 to 91.25
Copper	4 to 5	4.25 to 4.75
Antimony	4 to 5	4.25 to 4.75
Lead, max.	0.35	0.35
Iron, max.	0.08	0.08
Arsenic, max.	0.10	0.10
Bismuth, max.	0.08	0.08
Zinc and Aluminum	None	None

SPECIFICATION No. 11, BABBITT

Composition in percentage:

	No. 11 Cast Products	No. 11A Ingots
Tin	86.00 to 89.00	87.25 to 87.75
Copper	5.00 to 6.50	5.50 to 6.00
Antimony	6.00 to 7.50	6.50 to 7.00

General Information.—This is a rather hard babbitt which may be used for lining connecting-rod and shaft bearings which are subjected to heavy pressures; its "wiping" tendency is very slight. It is also suitable for die castings.

SPECIFICATION No. 12, BABBITT

Composition in percentage:

	No. 12 Cast Products	No. 12A Ingots
Antimony	9.50 to 11.50	10.25 to 10.75
Copper	2.25 to 3.75	2.75 to 3.25
Lead	24.00 to 26.00	24.75 to 25.25

General Information.—This is a relatively cheap babbitt and is intended for bearings subjected to moderate pressures. It is also suitable for die castings.

SPECIFICATION No. 13, BABBITT

Composition in percentage:

	No. 13 Cast Products	No. 13A Ingots
Tin	9.25 to 10.75	9.75 to 10.25
Antimony	14.00 to 16.00	14.75 to 15.25
Lead	74.00 to 76.00	74.75 to 75.25

General Information.—This is a cheap babbitt and serves successfully where the bearings are large and the service light. It should not be used as a substitute for a babbitt with a high tin content. It is also suitable for die castings.

SPECIFICATION No. 14, BABBITT

Composition in percentage:

	No. 14 Cast Products	No. 14A Ingots
Tin	9.25 to 10.75	9.75 to 10.25
Antimony	14.00 to 16.00	14.75 to 15.25
Lead	74.00 to 76.00	74.75 to 75.25

General Information.—This is a cheap babbitt and serves successfully where the bearings are large and the service light. It should not be used as a substitute for a babbitt with a high tin content. It is suitable for die castings.

STATIONARY-ENGINE DIVISION REPORT

T. C. Menges, *Chairman*	Associated Manufacturers Co.
L. F. Burger, *Vice-Chairman*	International Harvester Co.
H. G. Holmes	Novo Engine Co.
V. E. McMullen	Hercules Gas Engine Co.
I. J. Nelson	Nelson Bros.
O. A. Powell	Cushman Motor Works
L. W. Witry	Waterloo Gasoline Engine Co.

FLYWHEEL PULLEY LUGS

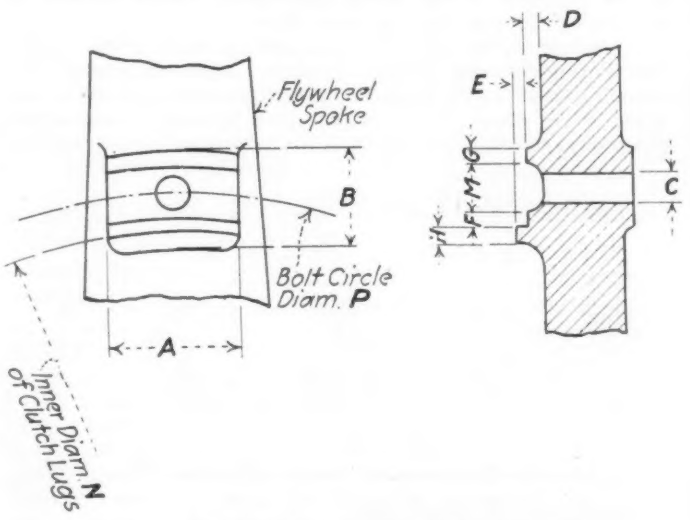
(Proposed S.A.E. Recommended Practice)

At the present time it is necessary for stationary internal-combustion engine builders to carry a large stock of pulleys in their branch houses to supply customers. This often necessitates carrying for 5 or 6 years in the distributors' stock pulleys for which there is practically no demand.

Appreciating this situation, L. F. Burger, of the International Harvester Co., was appointed a Subdivision of one at the February 1921 meeting of the Stationary-Engine Division to formulate a recommendation for flywheel pulley lugs. Mr. Burger submitted a recommendation that was subsequently extended at the suggestion of T. C. Menges, chairman of the Stationary-Engine Division, to include an additional size, the revised recommendation being approved by the members of the Division.

Information received in reply to a questionnaire sent to stationary-engine and pulley manufacturers showed that there is no general concordance of practice. This report should therefore be considered as an attempt to establish uniformity of pulley design to secure interchangeability and a minimum number of sizes. The manufacturers should have little trouble in eventually changing their product to the proposed dimensions. Once the standard is adopted in actual practice, each manufacturer will have to stock a much smaller number of pulleys. Therefore

The Division recommends that the flywheel pulley lug dimensions given in the accompanying table be adopted as S.A.E. Recommended Practice.



PULLEY AND FRICTION CLUTCH FASTENING LUGS

A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*M*	*N*	*P*	Number of Lugs to Pulley or Clutch
1½	1¼	9/16	¼	3/16	¼	¼	¼	¾	8½	9¾	6
1¾	1½	9/16	¼	3/16	¼	¼	¼	¾	14	15¼	6
2	2¼	9/16	¼	3/16	¼	¼	⅜	1⅜	16¼	17¾	6
2¼	2¼	9/16	¼	3/16	¼	¼	⅜	1⅜	18¼	19¾	6

PARTS AND FITTINGS DIVISION REPORT

Division Personnel

F. G. Whittington, *Chairman*	Stewart-Warner Speedometer Co.
W. C. Keys, *Vice-Chairman*	Gabriel Mfg. Co.
Clarence Carson	Dodge Bros.
E. R. Douglas	Cincinnati Ball Crank Co.

H. B. Garman	Steel Products Co.
H. S. Jandus	C. G. Spring Co.
F. W. Slack	Peerless Motor Car Co.
C. W. Spicer	Spicer Mfg. Corporation
Alex Taub	General Motors Corporation
E. W. Weaver	Weaver & Kemble Co.

Passenger-Car Front Bumpers

(Proposed Extension of S.A.E. Standard)

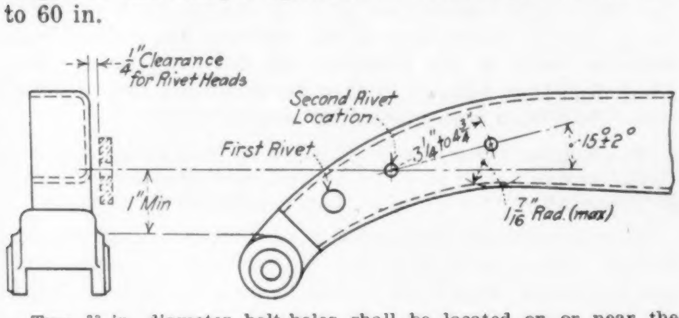
In 1921 a questionnaire was sent to passenger-car builders in the matter of the feasibility of standardizing a bolted-on front-bumper connection for passenger cars. Although it was pointed out that the adoption of such a standard would necessitate providing holes in the passenger-car frames, for use either by the car builders for mounting bumpers as standard equipment or by the car-owners for mounting bumpers as accessories, over 90 per cent of the companies replying recommended affirmative action. A Subdivision, consisting of F. G. Whittington, of the Stewart-Warner Speedometer Co., and E. W. Weaver, of Weaver and Kemble, was appointed to formulate a tentative recommendation.

As the bumper mounting is influenced largely by the frame design, a joint meeting of the Parts and Fittings and the Frames Divisions was held in November 1921, at which a report formulated by the Subdivision was discussed. The report was subsequently referred to passenger-car, bumper and rebound shock-absorber manufacturers for comment, and published in the April issue of The Journal.

The replies received, which were in general favorable, were summarized, referred to the members of the Division and discussed at the Division meeting in April. The report was revised at this meeting in certain details to make it conform as closely as possible with certain changes that were suggested. Therefore

> The Division recommends that the passenger-car front-bumper mounting shown in the accompanying illustration be adopted as an extension of the present S.A.E. Recommended Practice for Passenger-Car Front Bumpers, page C55 of the S.A.E. Handbook

The present S.A.E. Standard for Automobile Bumper-Mountings specifies that the distance from the center of the bumper face to the ground shall be 21 ft. for front bumpers and 22 in. for rear bumpers, the width of the bumper face, which shall be flat, to be 2 in. for front bumpers and 2½ in. for rear bumpers. The overall length of the bumper specified in both cases is from 59 to 60 in.



Two 11/16-in. diameter bolt-holes shall be located on or near the neutral axis of the frame section.

The first bolt-hole back from the spring eye may coincide with the first or second rivet hole.

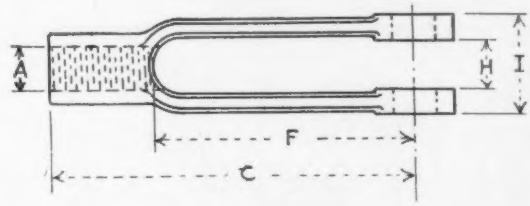
In cases where the second bolt-hole is to be used for mounting a shock absorbing device, the hole shall be located not more than 1 7/16 in. from the bottom of the frame channel at the nearest point.

Rod-Ends

(Proposed Extension of S.A.E. Standard)

Several communications were received by the Society to the effect that it was impossible to obtain ½-in. adjustable S.A.E. Standard rod-ends on the market with an adjusting length of 4 3/16 in. as specified in the present S.A.E. Standard on page C8 of the S.A.E. Handbook. It appears upon inquiry that the relative number of ½-in. adjustable yoke rod-ends 4 3/16 in. long sold at the present time is very small, the dimension generally used being 3 in. as specified in the original rod-end standard formulated by the Association of Licensed Automobile Manufacturers and adopted by the Society in 1911. The Parts and Fittings Division found that the rod-ends as specified in the present standard up to ½-in. nominal size are of comparatively light proportions, while those above the ½-in. nominal size are of comparatively heavy proportions. Therefore

> The Parts and Fittings Division recommends that the present S.A.E. Standard for Rod-Ends, page C8 of the S.A.E. Handbook, be extended by the addition of a ½-in. rod-end with a length of 3 in., the sizes to be classified as in the accompanying table.



LIGHT SERIES

A	C	F
3/16—32	1 9/16	1
¼—28	2	1¼
5/16—24	2¼	1 7/16
⅜—24	2½	1⅝
7/16—20	2⅞	1⅞
½—20	3	1⅞

HEAVY SERIES

A	C	F
½—20	4 3/16	3 1/16
⅝—18	4 15/16	3 11/16
¾—16	6⅛	4 9/16
⅞—14	7⅛	5¼
1 —14	8	6

Note: All other dimensions are the same as given in the table on page C9 of the S.A.E. Handbook.

Plain Steel Washers

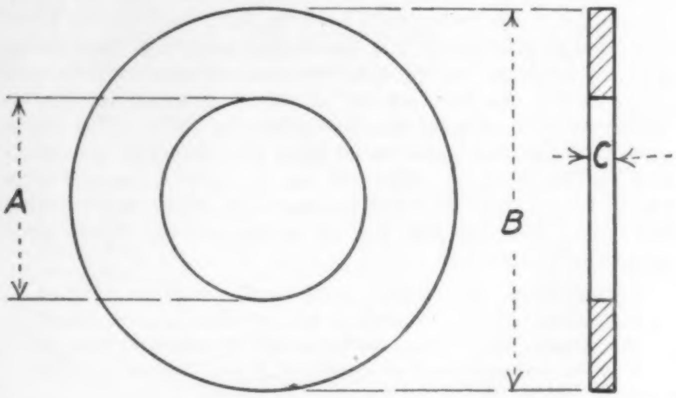
(Proposed S.A.E. Standard)

Last fall the Parts and Fittings Division was requested to prepare a standard for plain steel washers. Inasmuch as practice did not follow any established standard, a Subdivision was appointed to investigate the situation and prepare a report to the Division. At the last meeting of the Division data were submitted showing the die equipment in the possession of a number of washer manufacturers that conformed to a series of plain-washer sizes submitted to the Division and approved. The proposed series of sizes is intended for S.A.E. bolt sizes from ¼ to 1½-in. diameter and includes outside diameters that are suitable for use with U. S. standard hexagon nuts. Therefore

> The Parts and Fittings Division recommends the accompanying table of plain steel washer size for adoption as S.A.E. Standard

The chart accompanying this report shows the progression of S. A. E. bolt sizes, the plain washer inside

diameters, the distance across S.A.E. Standard hexagon bolt-heads and nuts and the outside diameter of the proposed washers.



Bolt. Diam.	WASHERS		
	A	B	C[3] ±0.010
1/4	9/32	5/8	1/16
5/16	11/32	11/16	1/16
3/8	13/32	13/16	1/16
7/16	15/32	15/16	1/16
1/2	17/32	1 1/16	3/32
9/16	19/32	1 3/16	3/32
5/8	21/32	1 5/16	3/32
11/16	23/32	1 3/8	3/32
3/4	13/16	1 1/2	1/8
7/8	15/16	1 3/4	1/8
1	1 1/16	2	1/8
1 1/8	1 3/16	2 1/4	1/8
1 1/4	1 5/16	2 1/2	5/32
1 3/8	1 7/16	2 3/4	5/32
1 1/2	1 9/16	3	5/32

[3] This dimension permits use of scrap stock. Washers shall be flat and free from burrs.

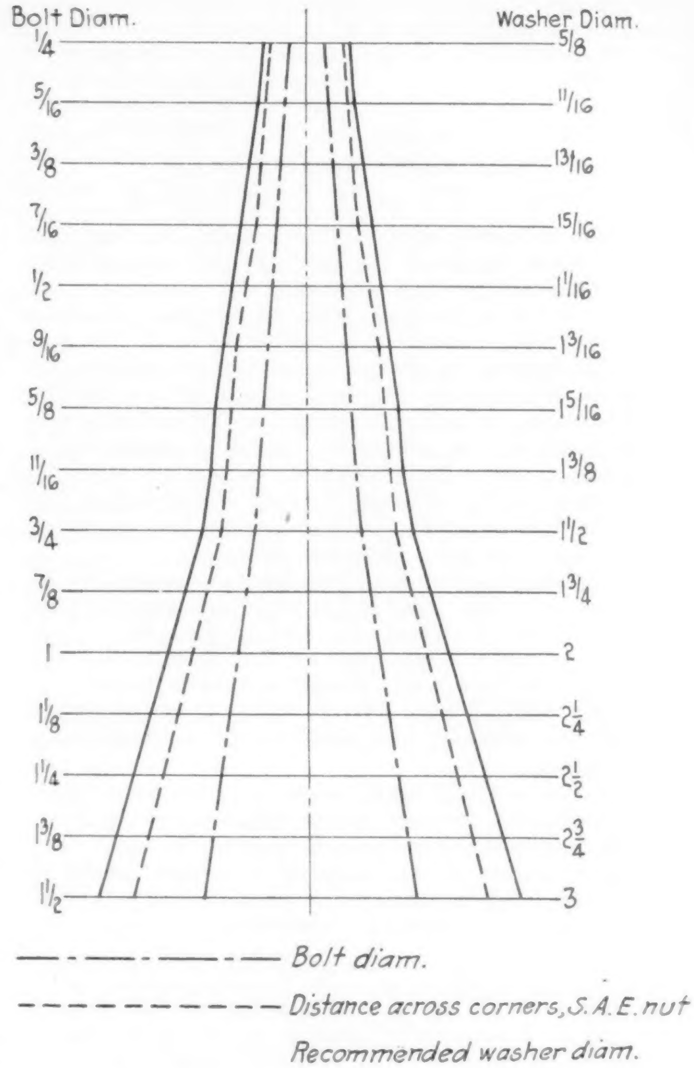
Ball Studs

(Proposed S.A.E. Recommended Practice)

In February 1919 a discussion was started of the Society standardizing ball-and-socket joints for reach-rods. The subject was assigned to the Miscellaneous Division and, after preliminary consideration, it developed into a study of current practice with regard to ball studs. Early in 1920 a large manufacturer of this product became interested and urged the adoption of standards as soon as possible. A Subdivision was then appointed, comprising

W. R. Strickland, *Chairman* — Peerless Motor Car Co.
E. E. Sweet — Lincoln Motor Co.

A tentative proposal was drafted and circulated and, in view of criticisms received, the work was reviewed and subsequently placed in the hands of a new subdivision, as conditions had arisen that prevented the original



subdivision from continuing its work. The new subdivision consisted of

F. W. Slack, *Chairman* — Peerless Motor Car Co.
E. R. Douglas — Cincinnati Ball Crank Co.

A new report was prepared and distributed, following which the Division's recommendation was submitted to the Standards Committee at West Baden in May 1921. As a result of discussion at this meeting the subject was referred back to the Division for further study. Mr. Slack was then obliged to give up this work and another new subdivision was appointed, constituted of

E. R. Douglas, *Chairman* — Cincinnati Ball Crank Co.
H. G. Garman — Detroit Steel Products Co.
W. J. Outcalt — General Motors Corporation

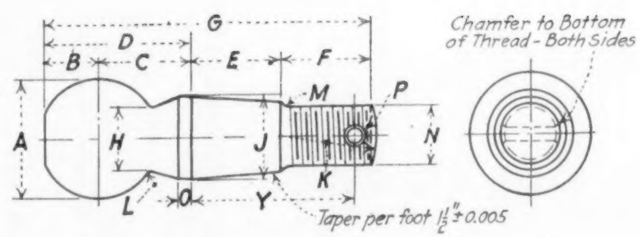
This Subdivision submitted a new report based on the earlier recommendations and the criticsims made, and was favorably considered by the Parts and Fittings Division at a recent meeting. Therefore

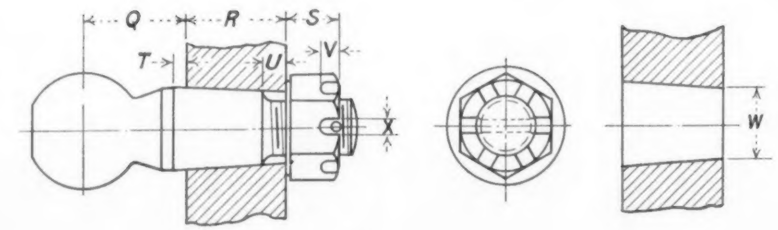
> The Parts and Fittings Division recommends that the dimensions for ball studs specified herewith be adopted as S.A.E. Recommended Practice

Serrated Shaft Fittings

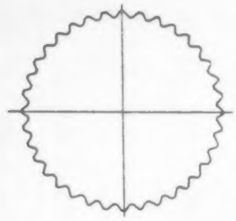
(Proposed S.A.E. Recommended Practice)

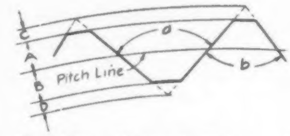
This subject was advanced for standardization in March 1920 by one of the large automotive parts manufacturing companies. The Parts and Fittings Division





A Nom.	A Actual	B	C	D	E	F	G	H	J	K	L	M	N S.A.E. Thr'd.	O	P Drill No.	Q	R max.	R min.	S	T	U	V	W	X	Y
½	31/64 ±0.005	7/32	7/16	21/32	7/16	½	1 19/32	9/32	13/32 ±0.002	⅜	1/32	1/32	5/16—24	3/32	48	9/16 +0 −3/64	19/32	33/64	17/64	⅛ +0 −3/64	13/64 +0 −3/64	3/32	0.329 ±0.001	5/64	27/32
⅝	39/64 ±0.005	9/32	9/16	27/32	9/16	½	1 29/32	11/32	15/32 ±0.002	½	3/64	1/32	5/16—24	3/32	48	11/16 +0 −3/64	23/32	41/64	17/64	⅛ +0 −3/64	13/64 +0 −3/64	1/32	0.376 ±0.001	5/64	31/32
¾	47/64 ±0.005	5/16	⅝	15/16	⅝	9/16	2⅛	13/32	35/64 ±0.002	⅝	3/64	1/32	⅜—24	7/64	36	25/32 +0 −3/64	25/32	43/64	21/64	5/32 +0 −3/64	13/64 +0 −3/64	⅛	0.446 ±0.001	⅛	1 5/64
⅞	55/64 ±0.005	⅜	11/16	1 1/16	11/16	¾	2½	15/32	⅝ ±0.002	⅝	1/16	1/16	½—20	⅛	36	55/64 +0 −3/64	29/32	25/32	7/16	11/64 +0 −3/64	17/64 +0 −3/64	9/64	0.509 ±0.001	⅛	1 5/16
1	31/32 ±0.010	7/16	¾	1 3/16	¾	¾	2 11/16	9/16	45/64 ±0.002	¾	1/16	1/16	½—20	⅛	36	15/16 +0 −1/16	31/32	27/32	7/16	3/16 +0 −1/16	9/32 +0 −1/16	9/64	0.578 ±0.002	⅛	1⅜
1⅛	1 3/32 ±0.010	½	27/32	1 11/32	27/32	29/32	3 3/32	⅝	25/32 ±0.002	1	1/16	1/16	⅝—18	5/32	28	1 3/64 +0 −1/16	1 5/64	15/16	35/64	13/64 +0 −1/16	19/64 +0 −1/16	3/16	0.642 ±0.002	5/32	1 19/32
1¼	1 7/32 ±0.010	9/16	15/16	1½	15/16	29/32	3 11/32	11/16	⅞ ±0.002	1	3/32	1/16	⅝—18	5/32	28	1⅛ +0 −1/16	1 11/64	1 3/64	35/64	13/64 +0 −1/16	19/64 +0 −1/16	3/16	0.725 ±0.002	5/32	1 11/16
1½	1 15/32 ±0.010	⅝	1⅛	1¾	1⅛	1 1/16	3 15/16	13/16	1 1/32 ±0.002	1¼	3/32	1/16	¾—16	3/16	28	1 11/32 +0 −1/16	1⅜	1 7/32	21/32	7/32 +0 −1/16	5/16 +0 −1/16	7/32	0.855 ±0.002	5/32	2
1¾	1 23/32 ±0.010	¾	1⅜	2⅛	1⅜	1 7/32	4 23/32	15/16	1¼ ±0.002	1¼	⅛	1/16	⅞—14	13/64	28	1⅝ +0 −1/16	1 21/32	1 15/32	49/64	¼ +0 −1/16	11/32 +0 −1/16	¼	1.039 ±0.002	5/32	2 25/64
2	1 15/16 ±0.015	⅞	1½	2⅜	1½	1 11/32	5 7/32	1⅛	1⅜ ±0.003	1½	⅛	1/16	1 —14	7/32	28	1 25/32 +0 −5/64	1 25/32	1 37/64	⅞	9/32 +0 −5/64	23/64 +0 −5/64	¼	1.147 ±0.002	5/32	2⅝
2¼	2 3/16 ±0.015	1	1 11/16	2 11/16	1 11/16	1½	5⅞	1¼	1 9/16 ±0.003	2	⅛	3/32	1⅛—12	¼	11	1 63/64 +0 −5/64	2 1/32	1 13/16	63/64	19/64 +0 −5/64	27/64 +0 −5/64	5/16	1.304 ±0.002	7/32	2 15/16
2½	2 7/16 ±0.015	1⅛	1⅞	3	1⅞	1⅝	6½	1⅜	1¾ ±0.003	2	3/16	3/32	1¼—12	17/64	11	2 11/64 +0 −5/64	2 7/32	2	1 3/32	19/64 +0 −5/64	27/64 +0 −5/64	5/16	1.468 ±0.002	7/32	3 15/64
3	2 15/16 ±0.015	1¼	2¼	3½	2¼	1⅞	7⅝	1⅝	2 1/16 ±0.003	2½	3/16	3/32	1½—12	19/64	2	2 37/64 +0 −5/64	2 19/32	2 11/32	1 5/16	21/64 +0 −5/64	27/64 +0 −5/64	⅜	1.733 ±0.002	¼	3 53/64





A and B are Nominally Equal
A + B = Depth of Cut
C in Hole and D in Shaft may be made as desired by each individual manufacturer beyond min and max as specified in table
N = Number of Serrations

Nominal Diam.	Pitch Diam.		*N*	*a*, deg.	*b*, deg.	HOLE			SHAFT		
						Large Diam. Min.	Small Diam.		Outside Diam.		Inside Diam. Max.
	Max.	Min.					Max.	Min.	Max.	Min.	
⅛	0.122	0.120	36	90	80	0.125	0.118	0.117	0.124	0.123	0.116
3/16	0.182	0.180	36	90	80	0.187	0.176	0.175	0.186	0.185	0.174
¼	0.243	0.241	36	90	80	0.250	0.235	0.234	0.249	0.248	0.233
5/16	0.303	0.301	36	90	80	0.312	0.293	0.292	0.311	0.310	0.291
⅜	0.363	0.361	36	90	80	0.375	0.352	0.351	0.374	0.373	0.350
½	0.485	0.483	36	90	80	0.500	0.469	0.468	0.499	0.498	0.467
⅝	0.605	0.603	36	90	80	0.625	0.584	0.583	0.624	0.623	0.582
¾	0.733	0.731	48	90	82½	0.750	0.716	0.714	0.749	0.747	0.713
⅞	0.855	0.853	48	90	82½	0.875	0.835	0.833	0.874	0.872	0.832
1	0.977	0.975	48	90	82½	1.000	0.954	0.952	0.999	0.997	0.951
1⅛	1.098	1.096	48	90	82½	1.125	1.071	1.069	1.124	1.122	1.068
1¼	1.220	1.218	48	90	82½	1.250	1.190	1.188	1.249	1.247	1.187
1⅜	1.342	1.340	48	90	82½	1.375	1.309	1.307	1.374	1.372	1.306
1½	1.464	1.462	48	90	82½	1.500	1.428	1.426	1.499	1.497	1.425
1¾	1.708	1.706	48	90	82½	1.750	1.666	1.664	1.749	1.747	1.663
2	1.952	1.949	48	90	82½	2.000	1.904	1.902	1.999	1.997	1.901
2¼	2.196	2.193	48	90	82½	2.250	2.142	2.140	2.249	2.247	2.139
2½	2.440	2.437	48	90	82½	2.500	2.380	2.378	2.499	2.497	2.377
2¾	2.684	2.681	48	90	82½	2.750	2.618	2.616	2.749	2.747	2.615
3	2.928	2.925	48	90	82½	3.000	2.856	2.854	2.999	2.997	2.853

All dimensions in inches.

referred the matter to C. W. Spicer as a Subdivision to draft a report. A questionnaire was sent to 75 companies, most of which submitted data, which were tabulated and used by Mr. Spicer as a basis for his report. The investigation showed that both taper and straight serrated-shafts are used extensively, although the practice is better defined for the taper type. The Subdivision's report was discussed in detail by the Parts and Fittings Division at a meeting held recently. Sizes below the ½-in. taper fitting have not been included in the recommendation because they are used very little if at all. The number of serrations on the taper fittings is already practically standard, whereas the number of serrations on the straight shafts varies considerably. The Division has, however, selected the serrations on straight shafts as shown in the report as providing a suitable number of well proportioned serrations. Therefore

The Parts and Fittings Division recommends that the taper and straight shaft-serrations shown in the accompanying illustrations and tables be adopted as S.A.E. Recommended Practice for Serrated Shaft Fittings

Tank and Radiator Caps

(Proposed Extension of S.A.E. Recommended Practice)

Trouble has been experienced by the gasoline-pump manufacturers in providing suitable nozzles for pump equipment because of the lack of uniformity in the size

(Concluded on page 523)

Overhead Camshaft Passenger-Car Engines

By P. M. HELDT[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* DRAWINGS

THE gradual trend toward overhead valves in automobile engines, as indicated by an increase in their use on American cars from 6 per cent in 1914 to 31 per cent in 1922, has been accelerated, in the opinion of the author, by their successful application to aircraft engines and by the publicity given them by their almost universal adoption on racing machines. Tractor engines recently brought out show the advantage of this construction. Methods of operating valves in the cylinder-head; the advantages of the valve-in-the-head construction as regards the form of combustion space, engine cooling and high-speed operation; the reason for using an overhead camshaft to operate the valves on racing engines, the question of noisy operation and the possibility of having an overhead camshaft engine operate as quietly as one in which the camshaft is enclosed in the crankcase; the location of the drive in the various foreign engines of the overhead camshaft type; the silent operation that is possible with a rear drive; the use of chains and spur, helical, worm and spiral bevel gears for the camshaft drive, with the advantages and disadvantages of each method and descriptions of specific applications; and some radical designs of overhead camshaft drive and valve-actuating mechanism that have been developed abroad are among the topics discussed. The three methods of operating the valves: (*a*) directly through the action of cams on followers secured to the end of the valve-stem, (*b*) through the interposition of single-armed levers or adjusting blades between the cams and the valve-stems, and (*c*) by the use of tappet levers, are also outlined with particular reference to the specific applications of each. Numerous illustrations supplement the text.

FOR a number of years there has been a gradual trend toward the use of overhead valves in automobile engines. This movement probably was accelerated by the success of aircraft engines with this valve arrangement, as well as by the publicity given to this construction by its almost universal use on racing engines. That valves located directly in the cylinder-head are not merely a fad but possess practical advantages can be concluded from the fact that practically all the tractor engines brought out during recent years have the valves so located.

Valves in the cylinder-head can be operated either by tappet-rods extending up one side of the engine or directly from an overhead camshaft. A third method, which consists of arranging the valves at right angles to the cylinder axis and operating them from a camshaft in the crankcase through the intermediary of long, double-armed levers, is exemplified in the Duesenberg engine. This arrangement necessitates the provision of a valve-pocket on top of the cylinder-head, and in most respects, especially as regards the form of the combustion-chamber, resembles the L-head engine more than the true valve-in-head engine which has no valve-pockets.

The advantage of the valve-in-head engine over the L and T-head types is that the combustion space has a much more favorable form. It has less cooling area in proportion to the volume than either of the other types. Consequently the heat loss to the jacket is reduced, and the distances from the firing points to the most remote part of the firing chamber are smaller, so that the flame is propagated through the combustible charge in a shorter interval of time. This tends toward increased power and higher fuel economy. The decrease in the loss of heat to the water-jacket due to the simple form of the combustion-chamber also facilitates the problem of effective engine cooling. The combustion-chamber, the portion of the cylinder casting subjected to the highest temperatures, is more symmetrical and the cylinders therefore are less likely to be distorted by unequal expansion. There is also less likelihood of the presence of masses of metal in the upper part of the cylinder casting that would be difficult to cool, as well as of the formation of steam pockets.

While considerable emphasis is laid on the advantages of the valve-in-head construction from the standpoint of engine cooling, the majority of designers who have adopted the practice have done so because they expected increased engine power and reduced fuel-consumption. The inlet as well as the exhaust-valve passages can be made shorter and more direct than in any other form of cylinder, and the overhead-valve type of engine lends itself particularly well to high-speed operation. It is also claimed that there is less tendency to knock. In view of the greater ease of effectively cooling the combustion-chamber this claim does not sound altogether unreasonable, but no definite proof of it seems to be available.

The tendency toward the overhead-valve type of engines is reflected by the analyses of car specifications that are being made annually. In 1914 only 6 per cent of the cars listed had such valves, while in 1922 the percentage had increased to 31. This applies to American cars. In Europe the valve-in-head engine has made progress, particularly during the last year. Disregarding engines of the sleeve-valve type, 32 per cent of the European models listed in 1922 had overhead valves, as against 20 per cent in 1921. British companies seem to be in a state of transition, many of them offering cars this year with both valve-in-head and side-pocket-valve engines.

TAPPET-RODS AND OVERHEAD CAMSHAFTS

In most engines of the valve-in-head type used on passenger cars the valves are operated by tappet-rods, whereas in most racing and aircraft engines they are operated by overhead camshafts. It is interesting to analyze the reasons for this difference. The side-rod method of overhead-valve operation is the older, at least in the sense that it was the first to gain much popularity. In one way it is also the simpler and more natural, for all that is wanted is a reciprocating motion, and

[1] M.S.A.E.—Engineering editor, Class Journal Co., New York City.

what means of transmitting a plain reciprocating motion over a moderate distance could be simpler than a push-rod? However, when engines attain high speeds the problems arising from inertia effects must be considered, and in order to permit high-speed operation overhead valves are generally employed. Increased power output results directly from the increased operating speed and the gain in economy is also in part dependent upon it.

In racing and ultra high-speed engines the speed of the engine is limited by the inability of the valves to close promptly at extremely high speeds. The inertia of the reciprocating parts of the valve becomes so great that it is practically impossible to get the cam followers to follow the contour of the cams. The force necessary to close the valves that is furnished by the valve-springs becomes greater in direct proportion to the weight of the reciprocating parts of the valve. In one sense there is almost no limit to the force that can be obtained from a spring, but in practice, if the pressure of the valve-spring is increased, the stress in the material of the spring generally is increased also with a consequently greater risk of breakage of the spring. A very stiff spring puts great stresses upon the valve itself. In order to permit higher operating speeds it is generally a much better plan to reduce the weight of the reciprocating parts of the valve than to increase the stiffness of the springs. To reduce the stress on the valves and their springs at extremely high speeds, motion should be transmitted in the rotating form as close to the valves as possible and then converted into a reciprocating motion. This is the reason that racing engines are so frequently provided with overhead camshafts.

Another aspect of the valve problem that needs consideration, particularly in connection with its application to passenger-cars, is the requirement that engines shall operate without appreciable noise. One cause of noisy operation is the clearance that must be given the valves, particularly the exhaust-valves, when they are cold. The need for this clearance is due to the fact that as the engine heats up the exhaust-valve stem expands much more than the engine as a whole and must be given freedom in which to expand or the valve will not close; a condition that is generally followed by disastrous results. If the cams are directly over the valves the chances of unequal expansion and the consequent need for clearance are greatly reduced. In this connection the relative temperatures of the cylinder-block and the valve-rods at the side are factors. If the cylinders are air-cooled and the valve-rods are located so that they are not greatly influenced by the heat of the cylinders but remain relatively cool, it is conceivable that the cylinders will expand much more than the rods. This difference in expansion has the same effect as the expansion of the valve-stems, and additional clearance must be allowed the valves. On the other hand, if the engine is water-cooled and the valve-rods extend through enclosed spaces within the cylinder-block, the rods and the block will be of substantially the same temperature and there will be no difference in expansion. Noiseless operation is not important in racing and aircraft engines.

There seems to be a general impression that it is difficult to make overhead camshaft gears operate noiselessly. Just why this should be so, if it is so, is not immediately apparent. Noise in cam gearing depends on the one hand on the cam outline, which could be made the same as in engines with the camshaft in the crankcase; and on the other hand on the mass of the reciprocating parts, that, as already pointed out, can be, and usually is, considerably less than in a valve-in-head engine in which the valve-rods extend up the sides. It may be that the impression was created in the early days when overhead camshafts were exposed. In modern engines of this type not only the camshaft but the whole valve mechanism is enclosed, so that it should be as easy to render an engine of this construction noiseless as one with the camshaft in the crankcase.

If there is any justification for the belief, the reason probably is that there are generally more wearing contacts in overhead camshaft gearing than in the more conventional type. At each of the contacts wear will occur and the play or slackness in the camshaft operating train therefore will increase more rapidly than in the ordinary L-head engine. Every time the nose of a cam passes from under the cam follower there is a tendency, which is more pronounced in the case of a four-cylinder engine than in one with more cylinders, for the camshaft to jump ahead and take up the slack of the drive, thus creating noise.

In connection with the older engines with side tappet-rods the rapid wear of the tappet-lever bearings is unpleasantly remembered. The only means of lubrication of these tappet-levers was an oil-hole and the excessive wear was probably due not so much to the inadequacy of the lubricating means as to neglect to make use of them. Presumably with the newer engines of this type having wick or even force-feed lubrication this difficulty has been overcome. With some of the overhead camshaft engines tappet-levers are also used, but they are now always enclosed and therefore can be effectively lubricated. The difference is therefore not one of camshaft location but rather of valve actuation and details of construction. Another thing in favor of the overhead camshaft is that it generally leads to a very symmetrical engine with the accessories so located that they are quite accessible and with the sides of the engine comparatively free from obstructions.

One method of judging an overhead-valve engine is by the facility for removing the cylinder-head for cleaning out the carbon by scraping and for grinding the valves. In an engine with the camshaft in the crankcase, the amount of disassembling and reassembling necessary for these two jobs is usually the same, but in the case of an overhead camshaft engine, this does not hold true. In many designs the whole camshaft and overhead valve-operating gear must be taken apart before the valves can be ground in, and this, of course, is a rather serious objection.

From the standpoint of manufacturing cost the overhead camshaft is not a very attractive proposition. The provisions that must be made for the complete, oil-tight enclosure of the camshaft for lubrication, the enclosure of the drive between members, of which one is carried by the crankcase and another by the cylinder-head, and the lubrication of this drive, involve considerable expense. There are ways of producing overhead camshaft engines comparatively cheaply but these engines are likely to prove unsatisfactory.

The overhead camshaft engine has not yet made much headway in this Country, only four cars that are at all well known having such engines at present. In Europe, on the other hand, many of the leading makes now employ this type of engine. These include Lanchester, Napier, Wolseley, Straker-Squire and Leyland in England; Bugatti, Farman and Hispano in France; Ansaldo and Diatto in Italy and Mercedes and Szawe in Germany. As a passenger-car engine the overhead camshaft type is practically an after-war product, but motor-truck engines

thus equipped were used by several companies in Germany prior to the war. I assume that the four American overhead camshaft engines, the Wills-Sainte Claire, the Leach Biltwell, the Duesenberg and the Frontenac, are well known, as all have been described in the technical journals within the past several months; consequently, only overhead camshaft engines of foreign design will be considered.

Location of Drive

In connection with the drive the question of its location at the front or rear comes up. It is the almost universal practice to have the camshaft drive at the front and this location is practically standardized. It is well known that trouble has been experienced with camshaft drives in the last few years and there is suspicion that the fact that the drive is at the front, where it is subjected to the effects of torsional vibration of the crankshaft, has something to do with it. Since the introduction of engines with six and more cylinders running at high speeds a great deal of trouble has been caused by torsional vibration. Owing to the periodic impulses in a gasoline engine, the engine naturally tends to accelerate and decelerate periodically and the flywheel is provided to minimize these speed fluctuations. The greatest fluctuations of speed occur at the forward end of the crankshaft. If the natural period of vibration of the shaft happens to coincide with the periodicity of the impulses received by this end of the shaft, a periodic torsional rocking motion is superimposed upon the regular rotary motion of the shaft. After each explosion in one of the forward cylinders the crankshaft will jump ahead, only to snap back the next moment. This effect, of course, occurs only at certain critical speeds of the engine and in engines with exceedingly stiff crankshafts these speeds may never be reached in regular operation. Where a critical speed is reached in the regular operation of the engine it is obvious that it must be rather hard on the camshaft gears if they are located at the forward end.

Mercedes for many years has located the camshaft gears at the rear as shown in Fig. 1 and another German maker, Szawe, has recently followed this example. If there is any part of the crankshaft that rotates more uniformly than the rest, it is the part at the flywheel. This, therefore, would seem to be the best part from which to take the camshaft drive. The objection has been made that this location renders the camshaft gears

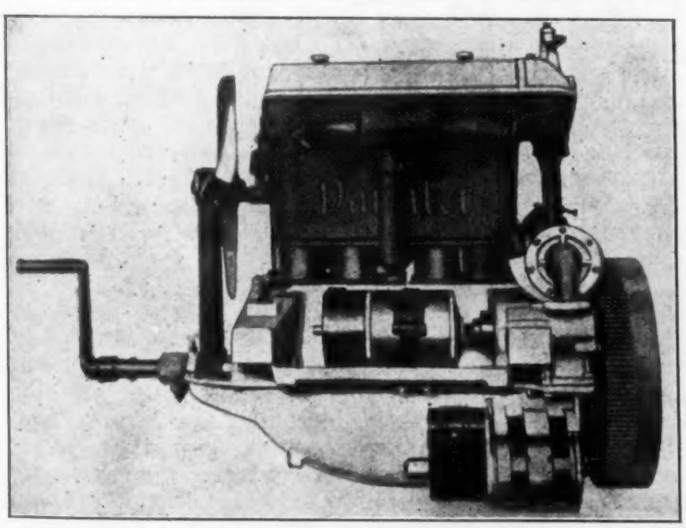


Fig. 1—Mercedes Overhead Camshaft Engine with Rear End Camshaft Drive

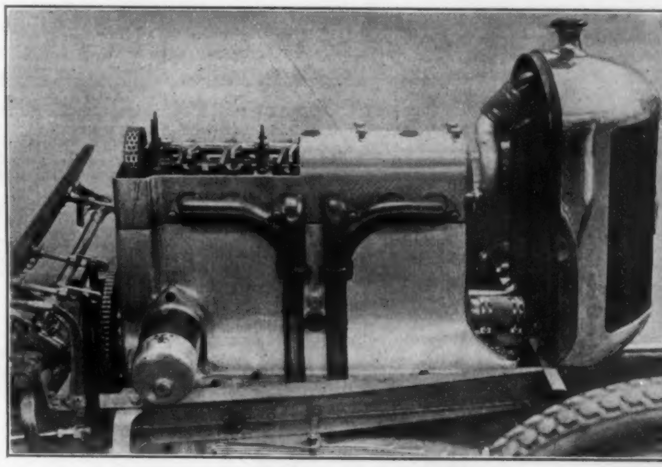


Fig. 2—British A. C. Engine with Chain-Driven Overhead Camshaft

much less accessible. This is true, but if it renders the gears at the same time invulnerable, what does it matter? With the light work the camshaft gears really have to do it should not be difficult to provide a set of gears that would last the life of the engine. In that case the question of accessibility would not arise.

I believe that anyone contemplating the production of an engine with an overhead camshaft should give the location of the camshaft driveshaft the fullest consideration, especially if the engine has six or more cylinders. From the standpoint of silent operation also the rear drive have something to commend it and conditions are still further improved when the fan is driven from the forward end of the camshaft, making the resulting or total torque on the camshaft non-reversible and thus eliminating the cause of noisy operation referred to in the foregoing.

Silent Chain and Spur Gear Drives

One method of driving an overhead camshaft is by silent chains and this has been used in a few cases for many years. One of the four American overhead camshaft engines, the Frontenac, has this form of drive. From the manufacturing standpoint the chain drive is very attractive, as apparently all that is required is a sprocket wheel on the crankshaft and another on the camshaft and a chain to connect the two. In practice however various difficulties crop up which have militated against the more extensive use of chains. A single chain between the crankshaft and camshaft is unsatisfactory because it would have to be abnormally long, and with the wearing of the links or the "stretching" of the chain it would whip and be likely to hit the wall of the casing, unless the latter was made with a large clearance space. Two chains therefore are generally used, one transmitting the motion to a short intermediate shaft, which may be the fan shaft, and the other transmitting it from this shaft to the camshaft. This does not dispose of the question of chain adjustment however. One way of solving this problem consists in driving not directly to the camshaft but to a short shaft at the end of it that drives the camshaft through a Hookham joint, and then providing both this short shaft and the intermediate or fan shaft with an eccentric adjustment. Another method consists of the use of chain-tightening idlers.

In the British A.C. six-cylinder engine the camshaft drive is by a long silent chain direct from the crankshaft to the camshaft, with a spring-loaded jockey pulley to take up the slack. The drive is at the rear end of the

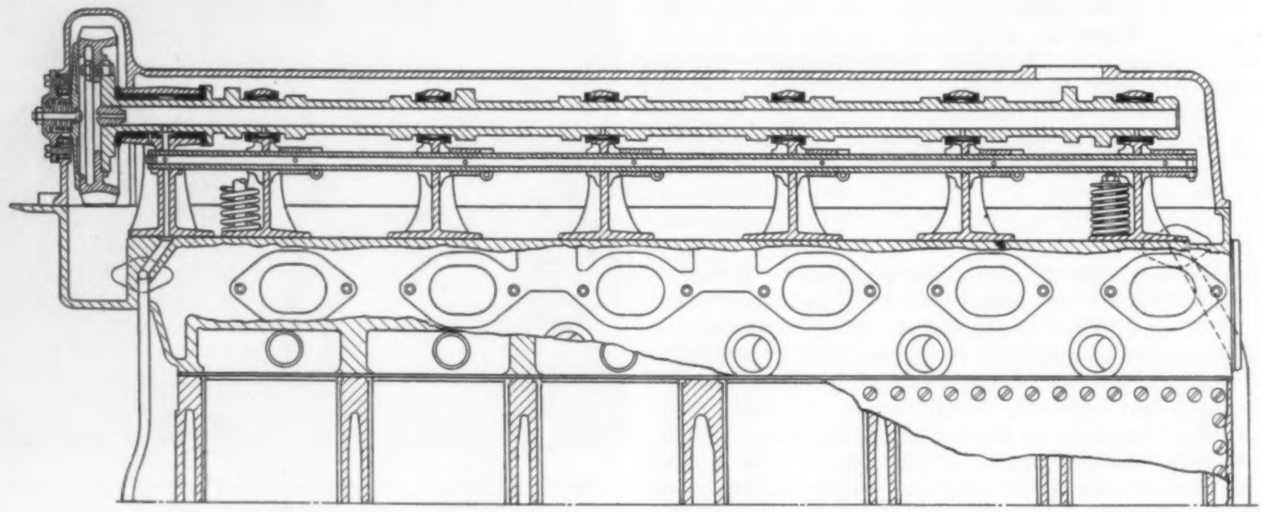


FIG. 3—THE UPPER PORTION OF THE NAPIER OVERHEAD CAMSHAFT ENGINE

engine, as is shown in Fig. 2. The ill-effects of the intermittent torque that the chain must withstand, even in this six-cylinder engine, are reduced in a degree by reason of the water-circulating pump being located at the end of, and driven directly by, the camshaft; the load represented by the pump being positive and uniform puts the torque always above the zero line, and though it does not eliminate or even reduce the torque variations due to cam-succession it maintains a positive pressure between the teeth of the sprockets and the chain. For this reason such a pump location seems to be worth considering, particularly on four-cylinder engines, no matter what form of drive is used.

On the Wolseley engine there is a short silent chain for transmitting power to an intermediate shaft directly above the crankshaft. Because of the shortness of this chain the troubles of whipping, and the like, are absent and no means for tightening the chain are required.

The spur-gear drive has given excellent service on racing engines. This involves a train of spur-gears extending up the front of the engine, and one would expect it to be rather difficult to make such a train run quietly, even if helical teeth were used. In racing engines the shafts of the intermediate gear are mounted on ball bearings, and this would no doubt also be required in high-class passenger-car engines, which would make the drive rather expensive. Another objection to the spur-gear train is that, since there are a number of wearing surfaces, when the gears begin to show serious wear a considerable backlash will soon develop. It would then probably be almost impossible to keep the drive quiet. The gears might be so liberally proportioned that the wear would be a negligible quantity, but the high cost and the rather unsightly appearance of the large gearcase would remain. An advantage shared by the two methods of drive is that they require no thrust bearings, whereas all the other more common forms of drive require such bearings. These involve considerable expense, including not only the cost of the bearings themselves but also that of machining the mountings and fitting the bearings.

A form of drive for overhead camshafts that has met with success and is now used on several well-known makes of cars is that through helical gears with axes at right angles, or the worm drive. Formerly the so-called

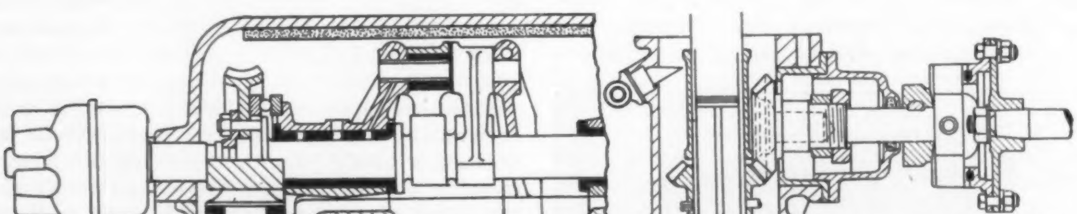


FIG. 4—CAMSHAFT DRIVE OF LANCHESTER ENGINE

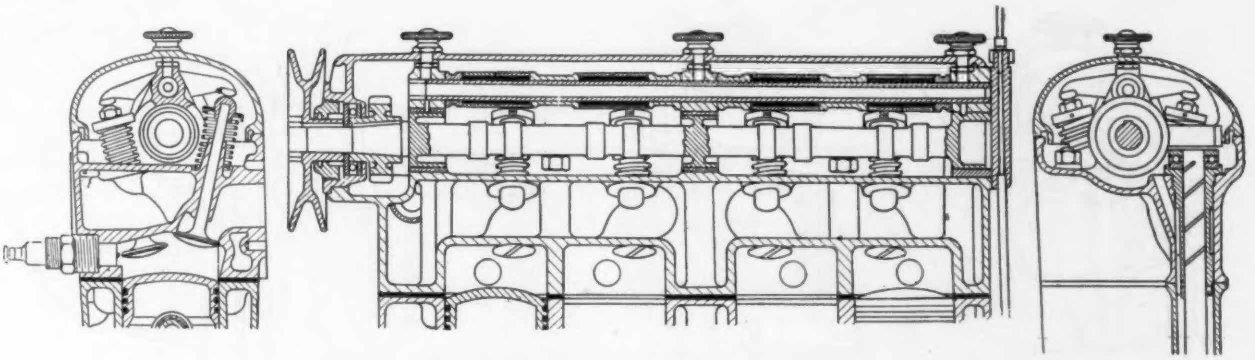


FIG. 5—DETAILS OF THE ANSALDO ENGINE
A Cross-Section of the Upper Portion Is Presented at the Left and the Middle Drawing Is a Side View of the Same Portion. The View at the Right Gives Details of the Camshaft Drive

spiral gears were used to a great extent for the camshaft drives of stationary gas engines. Their most valuable feature is their silent operation. If made of liberal size and provided with adequate means of lubrication they give satisfactory service, but as there is theoretically only point contact the unit pressure at the contact surface is quite high and if the lubrication is not dependable the backlash is likely to increase rapidly. For this reason several British manufacturers use true worm gears for the drive. Worm gears differ from helical gears in that the face of the wormwheel is throated, conforming to the circumference of the worm, whereby an increased bearing surface is obtained. One advantage of the helical gear or worm drive is that the vertical shaft is offset from both the crankshaft and the camshaft and therefore can have extensions for accessories coupled to it at both the top and the bottom. With a bevel-gear-driven intermediary shaft extra gearing is required for accessories at least at the bottom.

It is probably between the worm gear and the spiral bevel gear that the chief competition for supremacy in the overhead camshaft field will arise. The strong and the weak points of the two are practically the same. Both are absolutely quiet in operation, which constitutes their strong point, but both require means for the accurate adjustment of the mesh if they are to work satisfactorily. The advantage of worm gear in respect to greater handiness for the driving of accessories in certain locations is offset by its somewhat lower efficiency and lack of symmetry.

An overhead camshaft is one of the features of the six-cylinder aluminum engine brought out by Napier of England some years ago. The camshaft is hollow and is supported in seven cast brackets bolted to the aluminum cylinder-head as is shown in Fig. 3. It is located centrally above the cylinders. The valves are staggered, one being on one side of the camshaft and the other on the opposite side. There are two rocker-shafts supported on the same brackets as the camshaft. The rockers or valve-levers extend underneath the camshaft and are arranged so as to give a lift to the valves greater than the height of the cam nose. The camshaft drive on this engine is by worm gearing. The vertical shaft is made in three parts: one carried by the housing at the forward end of the crankcase; the second carried by the cylinder-head; and the third an intermediate shaft secured to the top and bottom parts by flanged couplings. From the bottom of the vertical shaft the oil-pump is driven, while the generator and magneto drive is taken off at the joint between the bottom and the intermediate section, the worm being combined with a coupling flange.

In connection with the Napier camshaft drive a description may be given of its camshaft brake, a feature found only on a few high-grade engines. A camshaft brake may be more necessary in an engine having a worm drive for the camshaft than in an ordinary L-head engine with direct spur-gear drive to the camshaft, owing to the fact that the worm drive might permit more backlash in the drive. The object of the camshaft brake is, of course, to prevent the camshaft from snapping ahead as the nose on any particular cam passes its follower. The brake consists of a metal disc mounted in the housing of the worm gear in such a manner that it cannot rotate but is forced against a flange on the rim of the worm gear by a coiled spring in a mounting hub bolted to the worm-gear housing from the outside. Adjustment is apparently provided for by washers under the flange of this hub.

Details of the camshaft drive on the Lanchester and Ansaldo engines are presented in Figs. 4 and 5.

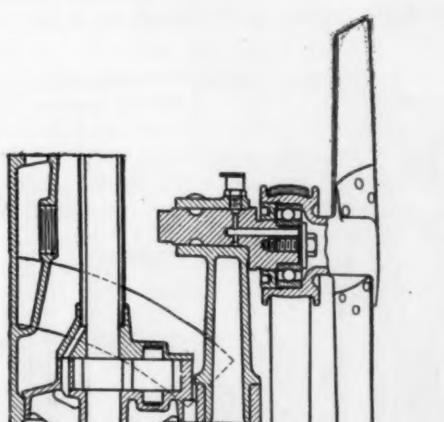


FIG. 6—OVERHEAD CAMSHAFT DRIVE ON THE WOLSELEY ENGINE

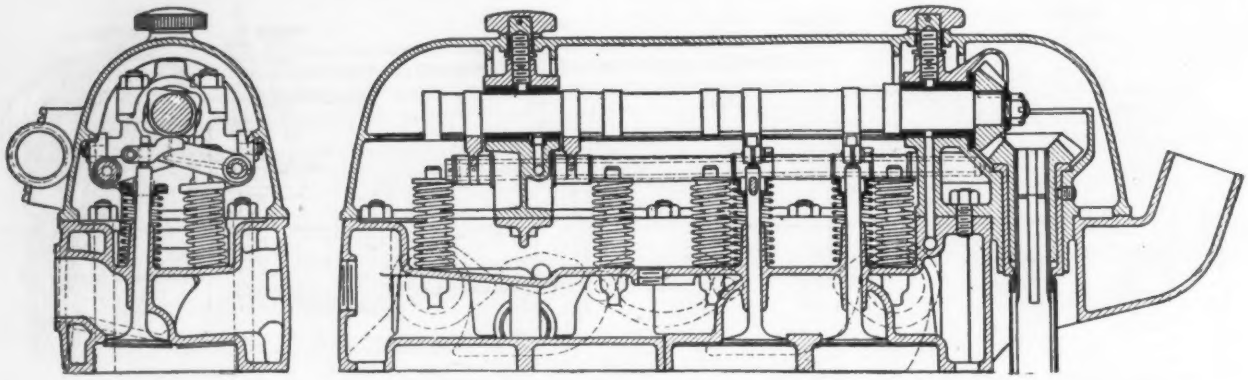


FIG. 7—THE UPPER PORTION OF THE WOLSELEY OVERHEAD CAMSHAFT ENGINE

SPIRAL BEVEL GEAR DRIVE

Probably the most prominent make of small car employing an overhead camshaft is the Wolseley. The construction which is shown in Figs. 6 and 7 is similar in some respects to that on the Napier engine but is simplified. The drive of the camshaft is by spiral bevel gear instead of worm gear. The spiral bevel gear drive is preceded by a silent chain drive from the crankshaft to a parallel shaft directly above it, at a ratio of 1 to 1 and with a very short chain. The 2 to 1 reduction is obtained by the lower pair of bevel gears which permits the use of a small pair of gears at the top, thus making for neatness. It would be practically impossible to obtain this reduction at the bottom of the vertical shaft and still have the driving pinion on the crankshaft, for the reason that to clear the crankshaft bearing the pinion must be of large diameter, and this would necessitate such a large-diameter driven gear that the whole drive would become unduly bulky. As it is, the fan-drive pulley is mounted on a forward extension

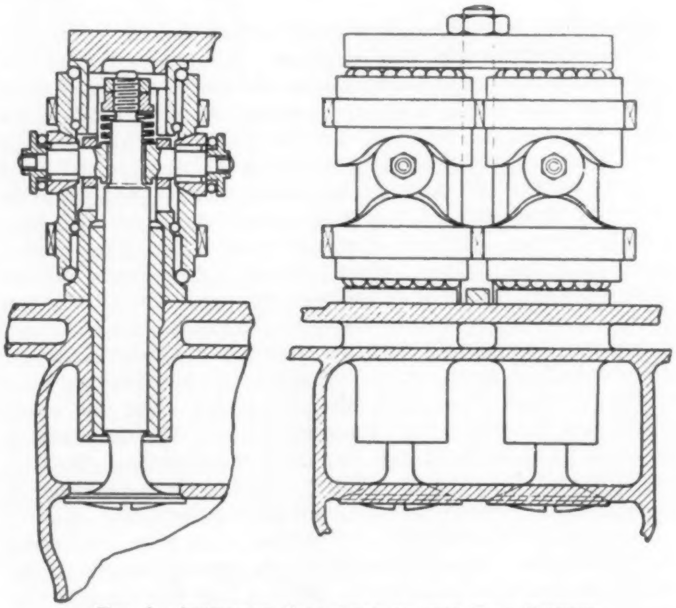


FIG. 9—OVERHEAD CAM GEAR ON THE FIAT ENGINE

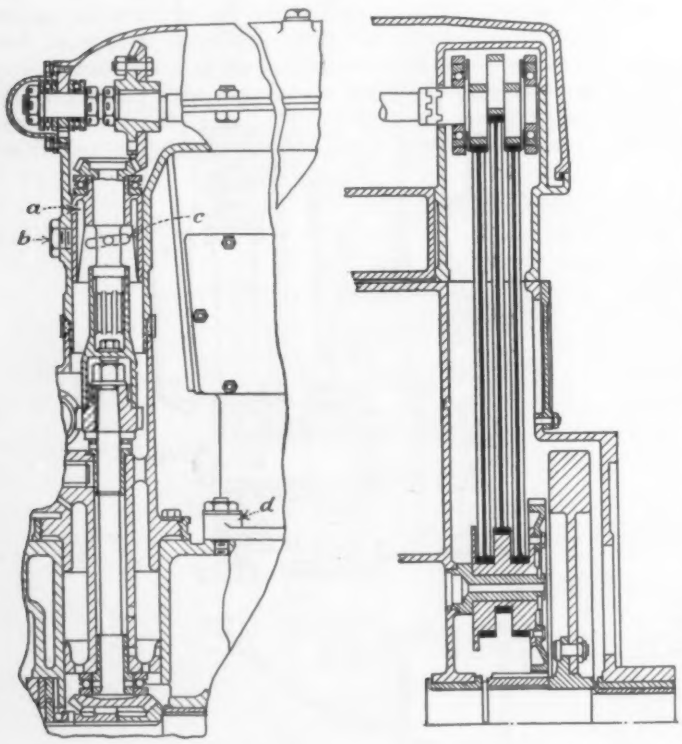


FIG. 8—AT THE LEFT THE CAMSHAFT DRIVE USED ON THE BENTLEY ENGINE AND AT THE RIGHT A SOMEWHAT UNUSUAL FORM OF CAMSHAFT DRIVE THAT IS USED ON THE LEYLAND STRAIGHT-EIGHT ENGINE

of the horizontal shaft and the gear type oil-pump forms part of the mounting of the vertical shaft. The shafts of this drive are mounted in plain bearings exclusively, except for a ball thrust-bearing on the horizontal shaft. This is a small-bore four-cylinder engine with the camshaft supported in two bearings only. To reduce the unsupported length as much as possible, the two rearmost cams are arranged to overhang the rear bearing. The general arrangement of the valves, rocker-levers and cams is practically the same as on the Napier. Both the camshaft and rocker-lever shafts are supported by brackets on the cylinder-head, the rocker levers being located underneath the camshaft and giving a greater lift to the valves than what might be called the cam throw. Clearance adjustment is made by set screws in the ends of the levers which are locked by clamping bolts in the split ends of the levers.

The spiral bevel-gear drive seems to have much to recommend it for overhead camshaft engines, but it must be realized that in order that the gears may run together quietly and efficiently they must be adjusted accurately. As there are two pairs of gears four adjustments are required and provision must be made for any lack of alignment or relative motion due to differences in heat expansion between the upper and lower bearings of the vertical shaft.

This problem has been attacked by two engineers of Bentley Motors, Ltd., an English company, that has taken out several patents covering a number of points of de-

sign. Referring to the left-hand portion of Fig. 8, the vertical shaft is made in two parts, which are joined by a telescopic splined coupling. Each part can be accurately adjusted endwise with or in its casing. At the bottom this is accomplished by providing the casing with a flange, which, instead of being bolted directly to the crankcase, is provided with a threaded ring. This ring can be screwed up or down over the integral flange and with it forms a flange of adjustable thickness. At the top a bearing box *a* can be rotated slightly by inserting a bar through the hole closed by plug *b*, and by being rotated is adjusted up or down by means of the helical slots *c* and pin screws in the casing extending into them. The camshaft is held against endwise motion by two thrust bearings at the forward end, and can be adjusted endwise by simultaneously turning the two nuts at the opposite sides of these bearings. It will be noted that the upper part of the casing for the vertical shaft is integral with the camshaft casing while the lower part is integral with the crankcase and the intermediate part bolts to this lower part. To permit the complete alignment of the upper and intermediate parts of the case the

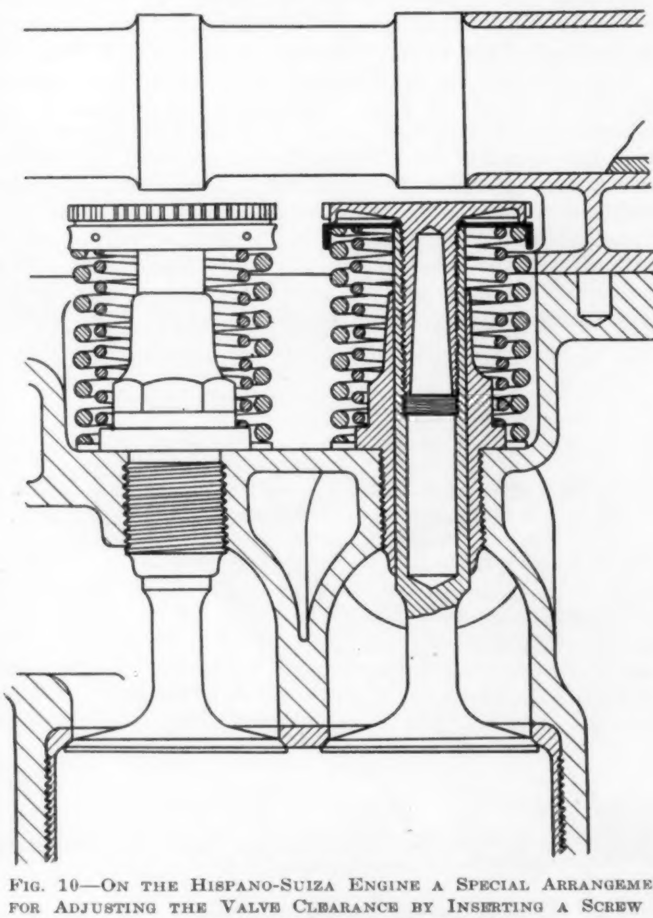


Fig. 10—On the Hispano-Suiza Engine a Special Arrangement for Adjusting the Valve Clearance by Inserting a Screw in the Valve-Stem Is Employed

bolt-holes through the cylinder-base lugs *d* are made oblong. The endwise freedom of the connecting-rods on the piston pins permits of slight variations in the position of the cylinder-block on the crankcase. It is thus possible to adjust the mesh of both sets of gears and also the alignment of the two parts of the vertical shaft housing.

Unusual Forms of Camshaft Drive

We now come to a number of rather radical designs of overhead camshaft drives and overhead valve-actuating

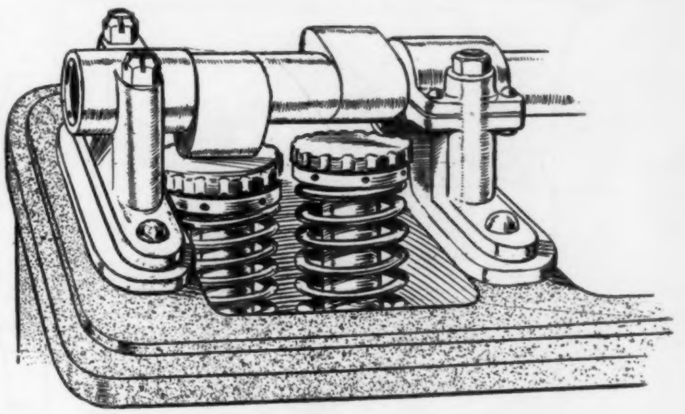


Fig. 11 — Sketch Showing the Relative Position of Valve Operating Cams That Act Directly on Followers Secured to the End of the Valve-Stem in the Hispano-Suiza Engine

mechanisms that have become known through recent patent publications. One of the cars that has attracted attention in the last year and a half is the Leyland eight-in-line. On the engine of this car an eccentric mechanism is used for transmitting motion from a half-speed shaft located close to the crankshaft and driven therefrom by a spur gear, to a short shaft normally in line with the camshaft. To obviate the difficulties due to the unequal expansion of the cylinder-block and the eccentric rods, the short shaft at the top has a support that is separate from the cylinder-head and is connected to the camshaft by an Oldham coupling which compensates for any slight misalignment between the driving and the driven members. The eccentrics are apparently set at angles of 120 deg. This drive ought to be commended from the standpoint of continuity. This construction is illustrated in Fig. 8.

A still more radical design of valve gear has recently been patented in England by the Fiat Co. Upon the upper part of the valve-stem is mounted a trunnion carrying two conical rollers. The trunnion is mounted between a shoulder on the valve-stem and a coiled spring that is backed up with a nut and lock nut at the top of the stem. Surrounding the valve-stem guide is a shell with opposite vertical slots through which the arms of the trunnion extend. On ball bearings upon this shell are mounted a pair of face cams upon the circumference of which are cut spur teeth. In the particular design

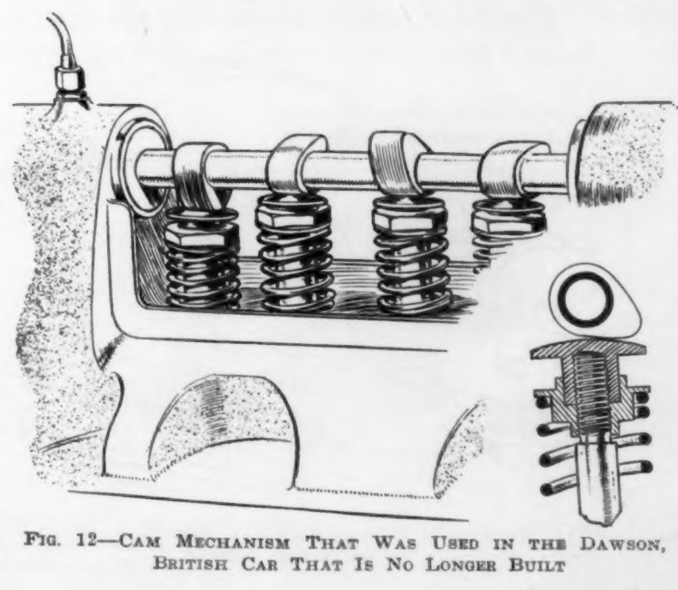


Fig. 12—Cam Mechanism That Was Used in the Dawson, a British Car That Is No Longer Built

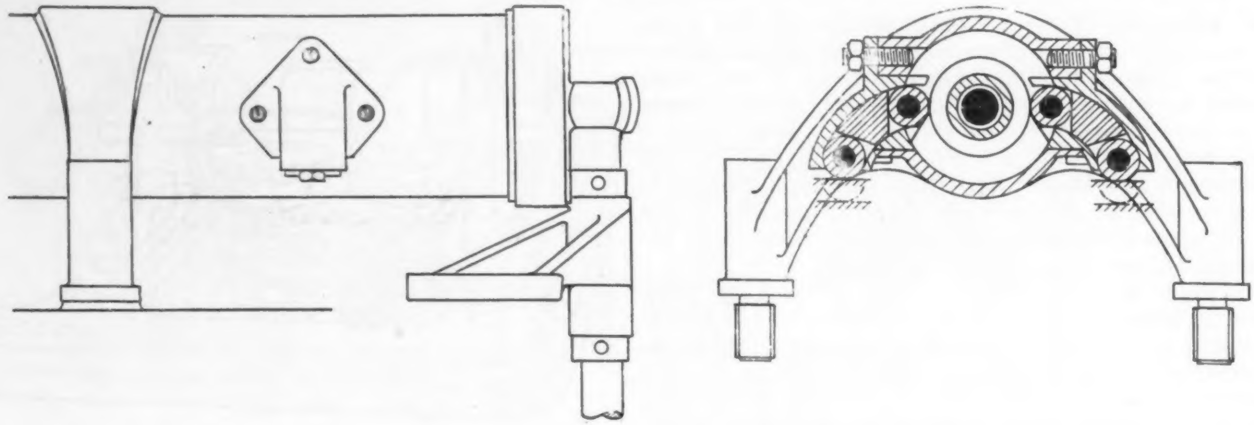


FIG. 13—THE BUGATTI OVERHEAD VALVE GEAR WHICH IS CHARACTERIZED BY CURVED QUADRANT-SHAPED PUSH-RODS WITH ROLLERS AT BOTH ENDS

shown in Fig. 9 there are four valves in the cylinder-head and all the cams are geared together. It is obvious that as the cams are revolved the valves are opened and closed positively, except for the slight play of the springs. A valve gear of this type should permit very high engine speeds, but the cost of manufacture is undoubtedly high.

VALVE MECHANISMS

At present three methods are in vogue for the operation of valves from overhead camshafts: The cams can act directly on cam followers secured to the end of the valve-stem, as in the Hispano-Suiza engine; single-armed levers or adjusting blades can be interposed between the cams and the valve stems; or the cams can act on the valves through the intermediary of tappet levers as in the Liberty aircraft engine. With the first-mentioned arrangement means for adjusting the valve clearance must be provided on the valve-stem itself. This problem was neatly solved by Birkigt, the designer of the Hispano engine. He drilled out the valve-stem and inserted a screw with a flat head with slotted rim. Underneath the screw head is a washer which has splines or keys on its inner circumference that engage into slots in the end of the drilled valve-stem, so that the washer can slide longitudinally on the valve-stem for a limited distance but cannot rotate with respect to it. The valve-spring presses the washer strongly against the screw head and the screw is thus locked. On the circumference of the washer there are several radial drill holes. A pin wrench is provided to engage into the holes and slots of the washer and screw head and permit the adjusting screw to be screwed farther into or out of the valve-stem. The details of this arrangement for adjusting the valve clearance are apparent from an inspection of Fig. 10, while the relative positions of the cams and the followers on the end of the valve-stems are brought out in Fig. 11.

Fig. 12 shows the system applied to the British Dawson car which is now no longer built. As will be seen, it resembles the Hispano in being of the direct-acting type and has similar means for the adjustment of valve-clearance. Despite the prevalent idea that Hispano claims a master patent on direct operation, Dawson continued to turn out engines with this type of valve system without serious interference for about two years. The failure of the firm was not ascribable in any way to the type of valve-gear used on the engines. The scope of the

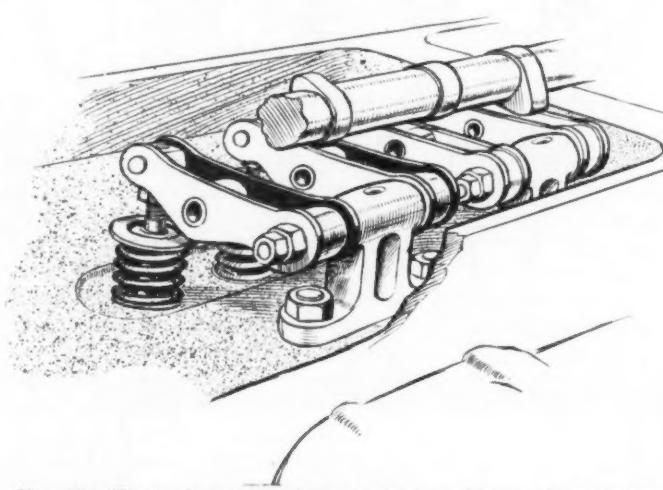


FIG. 15—VALVE OPERATING DETAILS OF THE 18-HP. FOUR-CYLINDER PHOENIX ENGINE

Birkigt Hispano patent apparently covers only the particular means of adjustment, including the pin wrench.

It is, of course, entirely possible to arrange overhead-valve mechanisms in a way similar to that used with the ordinary L-head engine. That is, between the valve and the camshaft can be placed a push-rod which is provided with an adjusting nut or adjusting screw and lock nut. The objection to this construction is that it makes the engine abnormally high. An overhead camshaft engine is much higher than an L-head engine in any case and if a push-rod is placed between the valves and the camshaft the height of the engine is increased by the length of the push-rod. There may be instances where this is of no

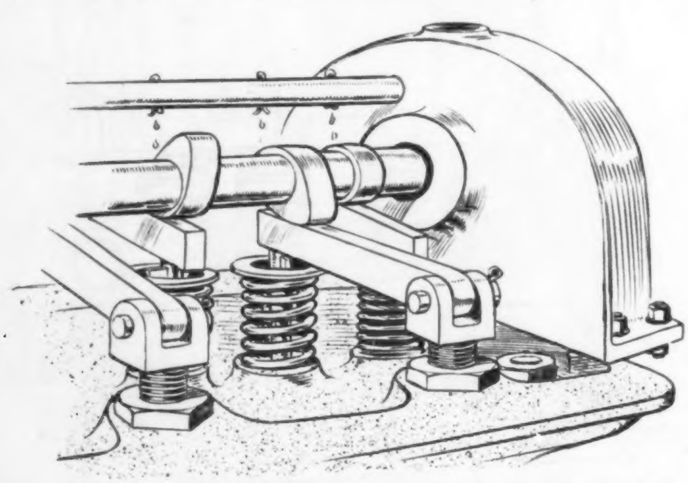


FIG. 14—CAM MECHANISM ON THE RHODE, A BRITISH LIGHT CAR HAVING A FOUR-CYLINDER ENGINE

great consequence as where a very small engine is placed under a normal sized hood, but in most cases it would be objectionable.

Bugatti developed an overhead valve gear which overcame this objection. He uses curved quadrant shaped push-rods with rollers at both ends. The cams on a camshaft placed centrally over the engine press against one of these rollers in a horizontal direction, while the other roller presses vertically down upon the valve-stem. The quadrant-shaped push-rods were located in guides bolted to the sides of the cylindrical camshaft housing with three bolts. The design illustrated in Fig. 13 is an early one, but the curved push-rod construction has been retained by Bugatti in a modified form. In this early design the provision for the adjustment of the clearance consisted apparently of the use of shims under the push-rod guide. Very frequently in recent designs the valves in the head are placed at an angle to the axis of the cylinder, one valve on each side of the central vertical plane of the engine, and the camshaft is then placed

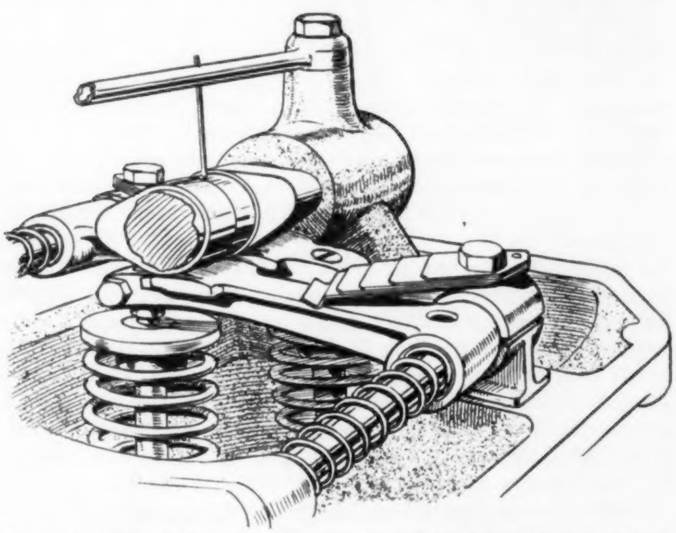


FIG. 16—THE BEARDMORE ENGINE THAT EMPLOYS PIVOTED LEVERS WITH AN OFFSET CAMSHAFT TO OPERATE THE VALVES

centrally over the cylinder-heads; an arrangement which makes for symmetry. On smaller engines the plan of setting the valves at an angle is not so practical, for it calls either for a wide cover or makes it necessary to leave the valve-springs unprotected and have the tappet-levers extend through the walls of the cover. In aircraft engines this latter practice is widely followed, because

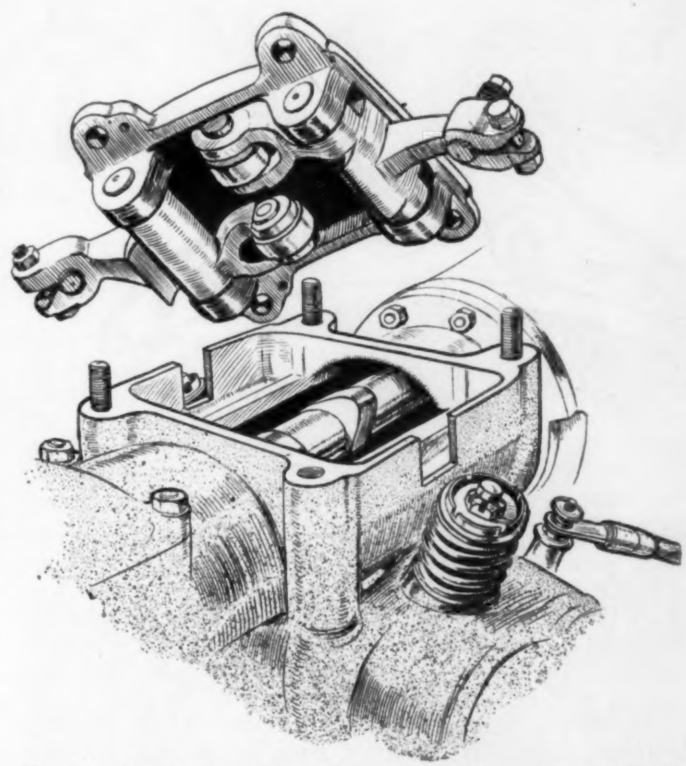


FIG. 18—THE OVERHEAD CAM MECHANISM ON THE STRAKER-SQUIRE SIX-CYLINDER ENGINE IN WHICH THE VALVES ON THE OPPOSITE SIDES ARE OUTSIDE THE CAMSHAFT CASING

dust and dirt are seldom factors to be considered in that line and the additional noise also is of no importance. In passenger-car engines vertical valves seem to be preferred and to make it possible to operate these valves from a central camshaft single-armed instead of double-armed tappet levers are used. The two valves are sometimes placed on opposite sides of the longitudinal central plane of the engine and are offset in the fore-and-aft direction sufficiently so that the two levers do not interfere. This arrangement permits the use of comparatively long lever arms that reduce the side pressure on the valve-stems and the enclosure of the entire mechanism in a compact housing.

One of the simplest, in fact crudest, overhead camshaft applications is that of the Rhode, which is shown in Fig. 14. This is a British light car with a four-cylinder engine of 67-cu. in. capacity, selling at a low price. Its camshaft, which is driven by a train of helical

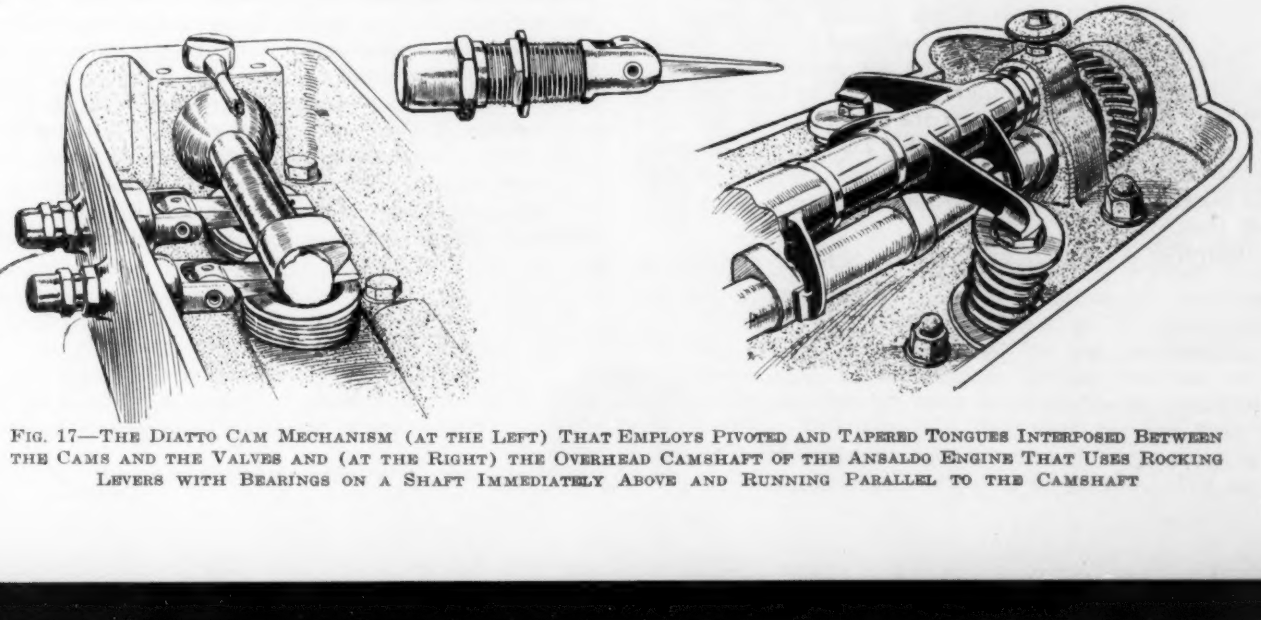


FIG. 17—THE DIATTO CAM MECHANISM (AT THE LEFT) THAT EMPLOYS PIVOTED AND TAPERED TONGUES INTERPOSED BETWEEN THE CAMS AND THE VALVES AND (AT THE RIGHT) THE OVERHEAD CAMSHAFT OF THE ANSALDO ENGINE THAT USES ROCKING LEVERS WITH BEARINGS ON A SHAFT IMMEDIATELY ABOVE AND RUNNING PARALLEL TO THE CAMSHAFT

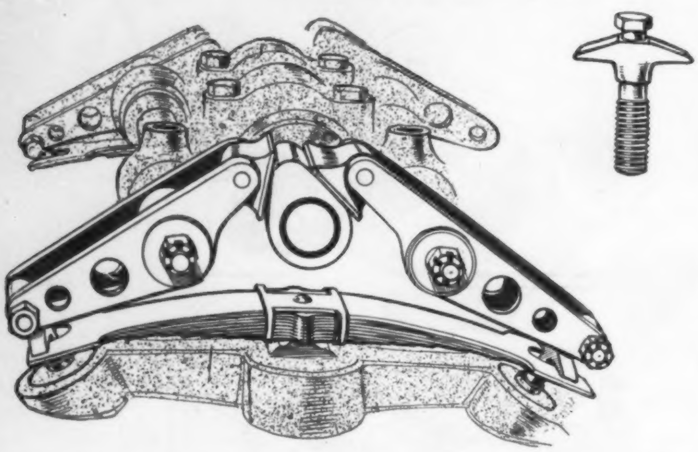


FIG. 19—THE OVERHEAD CAMSHAFT GEAR OF THE LEYLAND EIGHT-CYLINDER-IN-LINE ENGINE

pinions, actuates the valves through levers of rectangular stock pivoted on adjustable standards screwed into the cylinder-head and provided with a lock nut. The camshaft is offset from the valve-stem and the clearance adjustment is made by varying the height of the pivot pin centers. This implies first removing the pins and then raising or lowering the standards in half turns, a means of adjustment which is not very precise. As the drawing indicates, individual oil drips from an oil pipe running from one to the other of the pressure-fed camshaft bearings are provided for the cams.

A British chassis of moderate price, with a four-cylinder engine of 180-cu. in. capacity having an overhead camshaft, is the 18-hp. Phoenix, of which the valve-operating details are shown in Fig. 15. This also has pivoted levers interposed between the valves and an offset camshaft, but the levers in this case are built up and somewhat resemble the links of a roller chain; in fact, the cam makes contact with a roller on the intermediate cross-pin. Provision for clearance adjustment consists of cap nuts on the valve-stems; the lubrication system includes pressure feed to the shaft bearings and separate drips onto the cams from a longitudinal pipe not shown in the illustration. The drive is by helical gears and a vertical shaft having a dog coupling that is provided with offset jaws to ensure correct reengagement within limits.

The Beardmore engine, illustrated in Fig. 16, also comes within the pivoted-lever class with an offset camshaft, though in this case the levers for inlet and exhaust-valves respectively are pivoted on opposite sides of the cylinder-heads. The lubrication details are similar to those of the Phoenix. Small leaf-springs prevent chattering of the levers. Pairs of levers are separated on their pivot shaft by helical springs that take up end play and allow the levers to be moved to one side to facilitate valve removal and clearance adjustment. The means for doing this consists of threaded studs in the lever ends that make contact with the valve-stems and are locked by a pinch bolt.

A design of distinct interest is that of the Diatto, which is shown in Fig. 17 at the left. This has pivoted and tapered tongues interposed between cams and valves, the camshaft center being immediately over the valve-stem centers. The clearance adjustment is effected by moving the tongue toward or away from the camshaft center-line, the horizontal supports being threaded, located in clearance holes and locked by nuts in the wall of the overhead valve-chamber as shown. Without being crude this is a simple design that should lend itself to economical production far better than many overhead camshaft arrangements. The angularity of the tongues must be large at full valve-openings and would be liable to result in undue side thrust on the valve-stems, but there appears to be no valid reason why the tongues should not be longer and pivoted farther away from the valves to reduce the angularity and the side thrust. With efficient lubrication the wear of the tongues should not be excessive. In any case they could be made easily and cheaply renewable.

Typical of many other overhead camshaft designs, the Italian Ansaldo shown in the right half of Fig. 17 is in the class embodying rocking levers with bearings on a shaft immediately above and running parallel to the camshaft from end to end, with the valves on opposite sides of the cylinder-head. The valves are usually, as in the Ansaldo, outwardly inclined at a slight angle from the vertical. Although in the case illustrated the clearance adjustments are on the valve-stems, set screws and pinch bolts in the outer ends of the rockers are more in favor.

The Straker Squire arrangement, illustrated in Fig. 18, is unusual in two respects: In the first place it is used on a six-cylinder engine with separate cylinders; in the second place the valves on opposite sides are outside the camshaft casing. The drawing is practically self-explanatory, except that it does not show that the camshaft casing is a unit casting running from end to end of the cylinders and secured to them by the studs and nuts that hold in place the individual cover-plates that bear the roller-ended rockers. The drive in this case is by bevel gears at each end of a vertical shaft.

Still more unconventional is the Leyland eight-cylinder-in-line overhead camshaft gear, illustrated in Fig. 19, the drive for which already has been described. Only one cam is used for the inlet and exhaust-valves of each cylinder, the cam acting upon slipper followers pivoted to built-up rockers. These at their outer ends bear upon T-pieces screwed into the hollow valve-stems, while the springs are semi-elliptic laminated units forked at their ends to engage with and exert a lifting pressure on the T-pieces and through them upon the valves. With a total disregard for the cost of production, the Leyland at £10,500 for the chassis is the most expensive British car, the designer having evidently aimed at securing silent operation by endeavoring to prevent valve-spring rebound and so adopted the laminated type. The pivoted slipper followers may also have appeared to hold out possibilities in the same direction, but by being satisfied with the compromise valve-opening diagram that is obviously necessitated by using the same cam for inlet and exhaust, he would seem to have nullified any gain in efficiency that might have been produced by the overhead valves.

Automatic Charging of Motive-Power Storage-Batteries

By H. M. Beck[1]

Mid-West Section Paper *Illustrated with* Photographs and Diagrams

The paper deals with storage batteries suitable for the propulsion of vehicles. The service required is much more arduous and such batteries are of a much heavier type than that needed for starting service.

The necessity of proper storage-battery charging is emphasized and the elementary principles of the storage-battery are reviewed, a description following of the evolution of the modern storage-battery and its development with regard to commercial production and safety from mechanical and other injuries. Illustrations of this modern type of battery are presented and its advantages and characteristics enumerated and commented upon at some length, inclusive of difficulties that have been met and largely overcome.

Methods of automatic charge-control and cut-off are discussed and charts presented to show the results obtained through use of automatic devices such as the ampere-hour meter. The two-step method of charge is described and various applications of the storage-battery to modern usage are mentioned.

The title of this paper does not refer to starting batteries for internal-combustion engines. By motive-power storage-batteries we mean batteries used for the propulsion of any kind of vehicle. This includes batteries suitable for industrial trucks or tractors, the different kinds of storage-battery locomotive for use both in mines and for surface hauling and commercial street vehicles such as trucks and passenger-cars. The service required is much harder and the battery is of a much heavier type than is needed for starting an internal-combustion engine.

To be successful, a battery-propelled vehicle must be designed properly and be equipped with the correct type and size of battery but its success still depends absolutely upon the proper functioning of the battery. This seems to have been largely overlooked; at any rate, in the case of many battery-propelled vehicles very little attention has been paid to correct charging, upon which probably more than half of the success of the vehicle depends. Conditions vary in different types of service, and charging methods that are best for one type are not the best for another. After a very wide experience extending over many years, we have concluded that the full-automatic system of charging is the best and about the only practical one for motive-power service. We have tried the manual and the semi-automatic systems and find that there is no comparison with the full-automatic system in either the simplicity of maintenance or the results obtained. We do not expect any better results from the automatic system than we have obtained already from the manual in individual cases where the conditions happened to be right, but we do know from experience that with the former we can make the average results approach much more closely to the maximum.

The automatic charging of storage batteries is not a new idea; it has been used for many years in certain classes of service, but on a different basis. The practice has been to use it only where manual methods were impractical and to reproduce manual methods mechanically. The resultant apparatus is complicated and the service far from satisfactory. For example, automatic operation has been used for years for isolated gasoline-electric plants, and for steam-car lighting-equipment; more recently we have had the starting and lighting batteries on gasoline cars. The theory, however, has been largely to follow the manual method of charging at a constant rate, some automatic device being used to stop the charge at a specified voltage-limit. Although results of some value can be obtained, these methods are basically weak and on this account their use has been limited.

About 10 years ago we brought out a steam-car lighting-system that was fundamentally different from the rest. Instead of using a constant current with a voltage cut-off, we reversed the operation and applied a rather low constant voltage. This gives a taper charge due to the rise in the battery voltage as the charge progresses and the current tapers down almost to zero when the battery is fully charged. This system is one of the best examples of automatic charging to-day. It remains almost unchanged and produces excellent results but, unfortunately, its field of usefulness is somewhat limited.

The Pullman Co. brought out an improvement a few years later. This involved using its old car-lighting system and approximately a constant-current charge automatically controlled, and applying an ampere-hour meter cut-off in place of a fixed voltage cut-off. The system requires some adjustment and is complicated but, where the conditions are favorable, it approaches closely the results obtained from the constant-voltage system.

With the rapid growth of industrial electric trucks and locomotives a demand was created for the elimination of the charging problem, chiefly because of the class of labor involved. A thorough study of the problem was made and a system was laid out based on the inherent control characteristics of the storage battery, using the best experience of previous practice. It was found that not only a system, but equipment as well had to be developed. For example, although the ampere-hour meter was on the market, it was far from its present development. Suitable charging plugs, resistances and circuit-breakers were not available. The development has now reached a stage where we can cite actual practice.

Our automatic charge-control system has been in service in large fleets for approximately 4 years and has exceeded our expectations. We are no longer recommending the automatic system simply as a substitute for manual operation where that is not practicable, but rather in place of it because, through its use, we can obtain results that we have been unable to obtain in any other way. For example; the first battery that went out with the constant-voltage car-lighting system is still in operation after over 8 years of service. We can say

[1] M.S.A.E.—Engineer, operating department, Electric Storage Battery Co., Chicago.

that the plates could not have been in better condition if they had been operated under the best manual control. One of our first motive-power installations was in connection with two industrial-truck batteries, charged at night and cut-off by a night watchman when he happened to remember it. The result was that one battery was practically burned up within 6 months. We then installed the automatic system. This fleet now consists of nine trucks; the original batteries have been in use for 3½ years and are still giving good service. We installed a large fleet of electric locomotives equipped with what might be called a semi-automatic system. An automatic charge cut-off was provided, but the charging current was under hand control. Within less than a year we experienced trouble. The men supervising the battery charging were given overtime to complete the charge, whether that time was required or not. They found that by raising the rate the batteries became charged more quickly and they could get home that much sooner. The consequence was that the batteries were sadly overheated, with resulting injury. Arrangements were then made for automatic rate control as well as automatic cut-off. The battery temperature dropped 15 deg. in 1 month. Heat is one of the worst enemies of the storage-battery. These instances should give some idea of what the full-automatic system will accomplish and why we consider it the only practical system for this class of service.

Storage-Battery Principles

It will be easier to explain the development of the full-automatic charging-system if we first scan some of the general principles of the storage-battery. The first question that arises is why we use a storage-battery for the propulsion of vehicles instead of some other means. The storage-battery is one the simplest as well as most reliable pieces of apparatus available. Its very reliability is one of its weaknesses. The fact that a battery will operate for long periods after it needs attention results in its being often neglected until it is permanently injured. Most people think of a storage battery as being heavy, whereas, from an energy storage standpoint, it is relatively very light. A rubber band will store sufficient

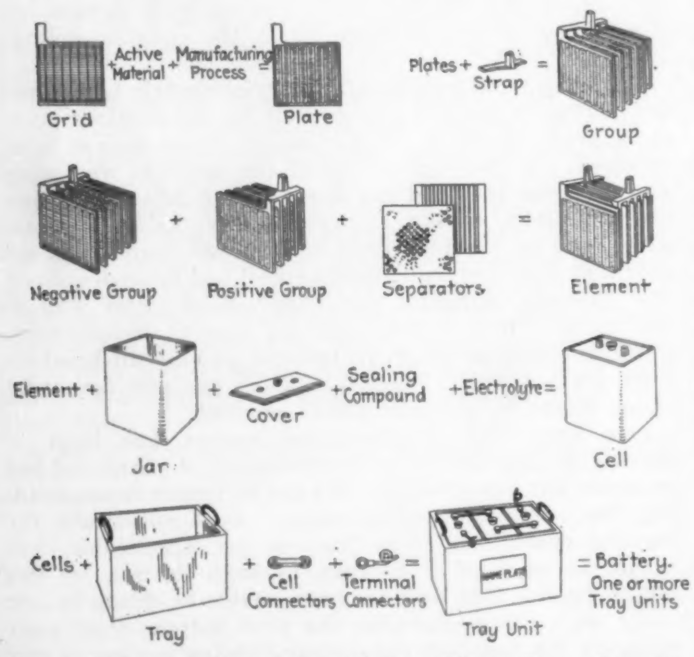


Fig. 1—Parts and Assembly of Typical Motive-Power Storage-Battery

energy to lift its own weight 300 ft.; and a steel spring about 500 ft. Compressed air, at 1000-lb. per sq. in. pressure, will lift its own weight, including the weight of the steel tanks, some 7000 ft.; whereas a storage battery will lift its own weight nearly 5 miles, about 3½ times the lift of its nearest competitor.

The storage battery was discovered by accident. About 1860 a Frenchman named Planté, while investigating the principles of electrolytic cells, submerged two strips of lead in a dilute solution of sulphuric acid and water and passed an electric current through them from an outside source. The two ends of the circuit accidently came together and he noticed that a meter in the circuit gave a reverse deflection, this indicating a discharge, and that he was getting back some of the current that had been put into the cell. He realized at once that he had discovered a means for storing electricity and from this simple model, unchanged in principle, we have the storage-battery of to-day.

The reason for the storage ability of this simple model is not difficult to state. Chemistry teaches that energy is always either released or absorbed when different elements combine, and that when the conditions are favorable this energy can be made to show itself in the form of an electric current. In Planté's cell we have two elements, sulphuric acid and lead. During discharge these combine, forming a salt of lead known as sulphate of lead. During charge, the lead sulphate is broken up, the acid is returned to the solution and the lead is brought back to its original condition. It should be noted that in this cycle nothing is lost, none of the chemical elements are used up and therefore nothing need be replaced. There is a general idea that the acid in a cell is used up during charge and discharge and must be replaced. During discharge, the elements combine and form lead sulphate and this combination releases energy in the form of an electrical current, but during charge the process is completely reversed, the combination is broken up and all of the acid is restored to the solution.

The Modern Storage-Battery

The modern storage-battery is not so simple as Planté's two pieces of lead immersed in a dilute solution of sulphuric acid and water, but the principle is identically the same and the development has been almost wholly mechanical. Fig. 1 shows the parts and assembly of a typical battery. A plate consists of a grid cast from lead alloy which acts as a mechanical holder for the finely divided lead that constitutes the active material. To obtain capacity the surface area of the plate must be large. We could accomplish this by simply making the plate very large, but it would soon become too big to handle conveniently; so we cut it into a number of sections and weld them to a cross-bar. This forms a group that is really a large plate cut into sections, acting otherwise as a single large plate. A positive group and a negative group assembled with separators for spacing and insulating the plates constitute a complete element. An element, a jar and cover, the sealing compound and the electrolyte form a complete cell; one or more cells, cell connections, terminal connections and trays constitute a complete battery.

The Ironclad positive plate which we are using in motive-power service is shown at the left of Fig. 2. In principle it is identical with a simple strip of lead, but it has been developed mechanically so that it will give far better service. The plate consists of a number of tubes joined by a metal bar at the top and bottom. The

tubes are made of hard rubber and are slotted or laminated horizontally to allow the electrolyte to circulate through the wall to the finely divided lead or active material with which they are filled. The finely divided lead surrounds a core of lead alloy that acts as the conductor. This plate is based originally upon a French design. The French plate, however, was impractical from a manufacturing standpoint and it has taken many years to develop the plate commercially. This design results in a maximum surface since the entire surface of the cylinders is exposed to the electrolyte and the rubber sheathing holds the active material in place. This greatly increases the life of the plate, which is roughly from two to three times that of the unprotected plate. The capacity also is greater and, due to its protection and rugged construction, this plate will withstand considerably more abuse than an equivalent flat plate. These characteristics indicate the selection of this plate for motive-power service, as the conditions are apt to be very severe.

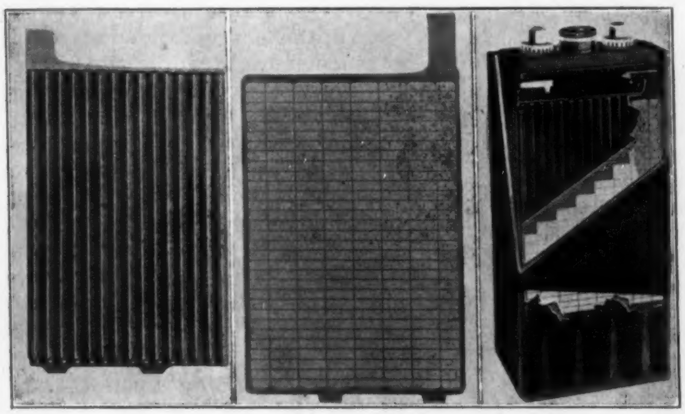


Fig. 2—At the Left the Positive Plate for a Motive-Power Storage-Battery; In the Center a Negative Plate and at the Right a Complete Cell

A negative plate, such as is shown in the center of Fig. 2, is not subject to the same action as a positive, and so does not require the same protection. On this account a flat plate has been selected and found perfectly practical for this cell. The two small feet at the bottom, also seen in the complete cell at the right, deserve attention. As will be noticed, the plates are supported on four bridges. The feet of the negative plates are offset as compared with the feet of the positive plates so that the negative plates rest on two of the bridges and the positive plates on the other two bridges. Thus, the positive and the negative plates do not rest on the same bridges and sediment collecting on top of the bridges can join only plates of like polarity and will not short-circuit the cell. These feet also permit the use of a thin wooden diaphragm, not over 1/16 in. thick and longer than the plates, as a separator. This prevents bridging of the material between the plates at the bottom, a trouble that sometimes occurs in the old construction.

In the early days of motive-power service, especially in mine locomotives where the mechanical conditions are probably more severe than in any other type of service, we had a great amount of jar breakage. This resulted in the development of the special so-called giant jar which is practically unbreakable in normal use. It was first used in mine locomotives but has since been adopted for nearly all types of motive-power service.

The two problems to be solved in automatic charging

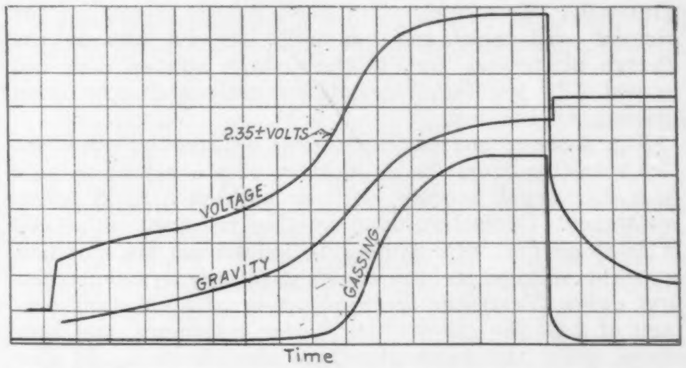


Fig. 3—Three Characteristic Curves Indicating the Voltage, Specific Gravity and Gassing of a Storage-Battery During Normal Charging at a Constant Rate

are, first, the control of the charging rate and, second, the cut-off. Neither one alone solves the problem and experience has shown that the full-automatic system is the only safe one for motive-power service. The rule for determining the proper charging rate is simple. As I have explained, we must avoid gassing and heating. These effects are closely related and if we avoid one we are apt to escape the other. Any system that avoids gassing and heating is safe so far as the battery is concerned, no matter how high or how low the charging current is.

Fig. 3 shows three characteristic curves, indicating the behavior of the voltage, specific gravity and gassing, during a normal charge at a constant rate. As will be noted, the voltage gradually increases as the charge progresses; toward the end the curve rises sharply, then bends over and finally reaches a maximum beyond which it will show no further increase no matter how long the charge is kept on. This maximum indicates the state of complete charge. The specific gravity behaves very much in the same manner. Gassing is very slight during the major part of the charge; there is a burst of gas near the end of the charge; the curve then tends to flatten out and reach a maximum. An interesting feature of the gassing curve, bearing on automatic control, is the fact that gassing in any considerable quantity is not shown below 2.35 volts per cell. From these curves we can deduce the following general principles. The voltage of a cell rises during charge and to charge at a constant current it is necessary to increase the applied voltage

Fig. 4—Curves Illustrating Different Characteristics Resulting from the Application of a Constant Voltage to a Discharged Battery With or Without a Fixed Series Resistance

gradually. Conversely, if a fixed voltage is applied the current will taper and gradually become less as the charge progresses and, if the voltage applied does not exceed 2.35 per cell, there is no serious danger from gassing or heating.

Fig. 4 shows curves illustrating different characteristics resulting from the application of a constant voltage to a discharged battery, with or without a fixed series resistance. Depending upon whether the fixed resistance is used or not, this system is known as the modified constant voltage, or the constant-voltage system. The first curve illustrates the application of a constant voltage of 2.25 per cell without series resistance, and this comes under the constant-voltage classification. It also represents the combination used in the case of the constant-voltage car-lighting system already mentioned. As will be seen, even at this voltage, which is considerably below that of the gassing point, 2.35 volts, the current starts at an extremely high value and then tapers rapidly to a very low value. It is due to this very low final current that no charge cut-off is required in the car-lighting system. The current starts at two or three times the normal rate if the battery is fully discharged, and tapers down to about 5 per cent of the normal rate when the battery is charged. While this particular system requires no cut-off, it is limited in its application because considerable time is required for a complete charge, on account of the low charging-rate during the latter part of the charge, and this time is too great for motive-power service. In car-lighting service, however, the conditions are different and this does not constitute very serious objection. In the case of a fleet of trucks the high starting rate is objectionable on account of the load factor. Therefore, to obtain a practical charging system, we must find some means to reduce the high starting-rate and raise the low finishing rate. We can raise the low finishing-rate by increasing the applied voltage. This will of course increase the already high starting-current, but this can be reduced by introducing a very low series resistance into the circuit; this resistance will be so low that it will have practically no effect upon the finishing rate. We thus have the so-called modified constant-voltage system.

The second curve, Fig. 4, is obtained by applying a

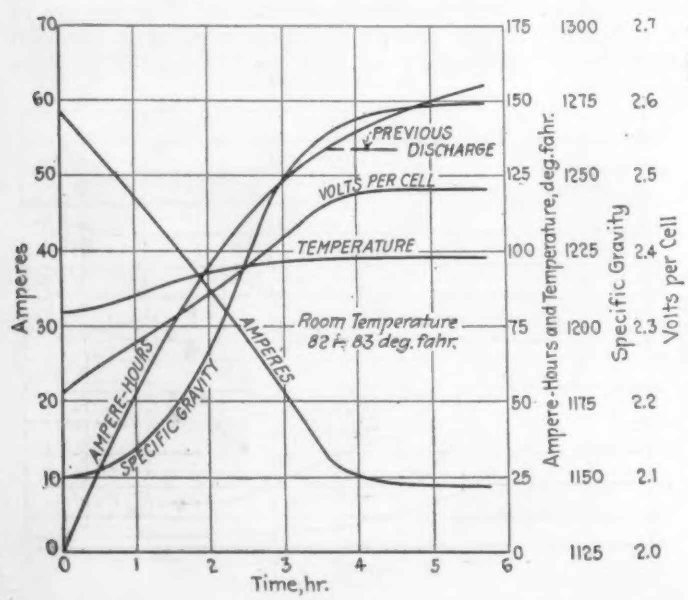


FIG. 5—CURVES OF ACTUAL READINGS TAKEN DURING A QUICK CHARGE REQUIRING ABOUT 5 HR.

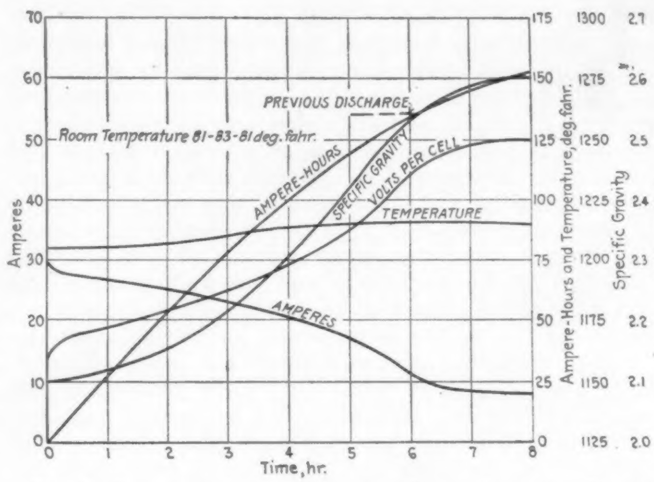


FIG. 6—CURVES OF ACTUAL READINGS TAKEN DURING A SLOWER CHARGE REQUIRING APPROXIMATELY 8 HR.

voltage of 2.53 with a series resistance of about 0.00306 ohm per cell. As is shown the current now starts at a lower point, holds up much more steadily, finishes at a higher point and the charge is completed in about 5 hr. under perfectly safe conditions; in fact, in the case of one fleet of locomotives that has been in service more than 2 years, only 4 hr. is available for the charge on account of the arrangement of shifts and, while this is an extreme case, it is being handled automatically.

The third curve, Fig. 4, shows the results obtained from a higher applied voltage, 2.63, and a higher series resistance of 0.0095 ohm. The curve starts at a still lower current value, ends at the finishing rate and the charging time is increased to 8 hr. It will thus be seen that with this simple system, depending upon the inherent characteristics of the cell, we can obtain automatically almost any value of charging current. This is the basis of our whole system so far as the rate control is concerned.

The curves in Figs. 5 and 6 show actual readings taken during charge, the first representing a quick charge requiring about 5 hr. and the other a slower charge requiring about 8 hr., together with the voltage and fixed resistances used in each case.

Another method of arriving at this same result is to design a generator with the proper characteristic to give battery-voltage and current curves similar to those shown in Figs. 5 and 6. We then get the same current characteristic without the use of the series resistance. This is the basis of the small motor-generator charging outfits with inherent taper-current characteristics that are so largely used. The proper characteristic is obtained either through a rather large armature reaction in the case of a shunt generator or by the use of an inverted series field, so, the voltage increases slightly as the load on the machine decreases. These motor-generator sets are applicable only to charging one battery, or two at the most, since a generator cannot be designed with the proper characteristic for simultaneously charging several batteries in different states of discharge. On this account the modified constant-voltage system generally is used for large fleets.

METHODS OF CUT-OFF

The question of the automatic-charge cut-off is not so simple. Fig. 7 shows curves giving the voltage and gravity characteristics during a constant-current charge. As already explained, the voltage curve rises gradually

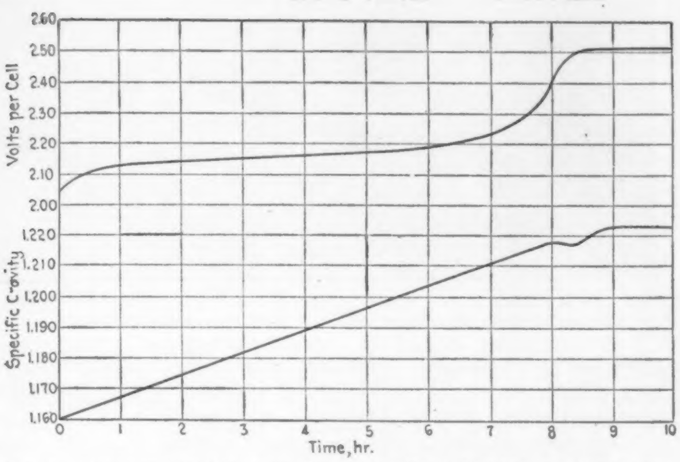


FIG. 7—VOLTAGE AND GRAVITY CHARACTERISTIC CURVES OF A STORAGE-BATTERY DURING CHARGING AT A CONSTANT CURRENT

during the bulk of the charge, much more rapidly just at the end and then bends over and reaches a maximum, which is the voltage indication of the full charge. The voltage was one of the first means employed as an automatic cut-off, but it is a difficult problem and has never been very successful. Any apparatus for showing when a maximum voltage is reached is complicated and would hardly be practicable in regular service. The actual voltage at which the cell is charged varies through wide limits. At high temperatures, the maximum reading may not reach 2.4 volts, whereas at extremely low temperature it has been known to go above 3.0 volts. There is no fixed voltage that will indicate when a battery is charged under varying temperatures. Further, the voltage depends directly upon the charging rate. If the charging current varies, the final voltage will vary. Finally, it depends upon the age of the cell and the strength of the electrolyte, and so many allowances are required that it has proved far from successful. The voltage cut-off has been used on a large scale in car lighting, but the so-called stop charges have been eliminated frequently because in many cases they interfered with rather than helped the operation.

The specific-gravity reading rises during charge much the same as the voltage and therefore can be used to cut-off the charge. This method has been employed in some types of stationary service, but it is decidedly difficult to get an accurate specific-gravity reading in motive-power assemblies, so that this method is almost out of the question for this service. The specific-gravity cut-off has most of the objections that apply to the voltage cut-off, its only advantage being that it is not so seriously affected by the charging rate.

The specific-gravity curve, Fig. 7, shows a slight irregularity near the end. This was formerly attributed to inaccuracy in the readings but, when recording hydrometers were developed, they showed it to be a regular characteristic. It is due to the rapid increase in the amount of gassing which occurs at this point. The combination of gas and electrolyte is lighter than solid electrolyte; so, the specific-gravity rise will be checked temporarily.

The ampere-hour efficiency of the cell is one of its most uniform characteristics. If 100 amp. hr. is taken out of a battery of given size, a close estimate can be made as to how many ampere-hours will be needed to charge it. This ratio is called the percentage of overcharge. It is not subject to many of the variations which apply to voltage or specific gravity and, within a small percentage, it is independent of temperature, the age of the battery or the charge and discharge rates.

THE AMPERE-HOUR METER

The ampere-hour-efficiency method depends upon some instrument for measuring the output and input in terms of ampere-hours. The ampere-hour meter is such an instrument. This is the instrument already mentioned as having been applied to the cars of the Pullman Co., and a decided improvement resulting. In fact, the addition of the ampere-hour meter has been responsible to a large extent for more than doubling the life of the batteries in this installation. The ampere-hour meter is shown in Fig. 8 at the left. It is simply a quantity meter and measures the product of current and time, or ampere-hours. The pointer moves clockwise when the current flows in one direction, and counterclockwise when the current flows in the opposite direction, thus registering charge and discharge. Further, it is designed so that it can be made to run at any given percentage slower in one direction than in the other. Hence, when the pointer gets back to zero it is indicated that a fixed percentage of overcharge has been put into the battery and that the charge is complete. The meters are equipped with a contact at zero that permits opening a shunt trip circuit-breaker and cutting off the charge when the zero point is reached. In other words, with this combination we have an automatic charge cut-off based upon the ampere-hour-efficiency principle which has proved the best if not the only practical method to date.

We have encountered prejudice against the use of ampere-hour meters, much of it being founded upon mistaken ideas. For example, there are several different grades or sizes of ampere-hour meter and the small cheaper type will not withstand motive-power service. In a very large percentage of cases where we trace the prejudice against ampere-hour meters we find it based upon trouble that resulted from the use of the wrong meter. An ampere-hour meter is a maintenance proposition. To take the position that because an ampere-hour meter fails it will not withstand the service, is not logical. An ampere-hour meter probably requires very much less maintenance than any other part of the equipment; this has been proved on a large scale. It does require some maintenance but is well worth this. One fleet of trucks in Chicago using about 80 of these meters sends in two or three meters per month for adjustment,

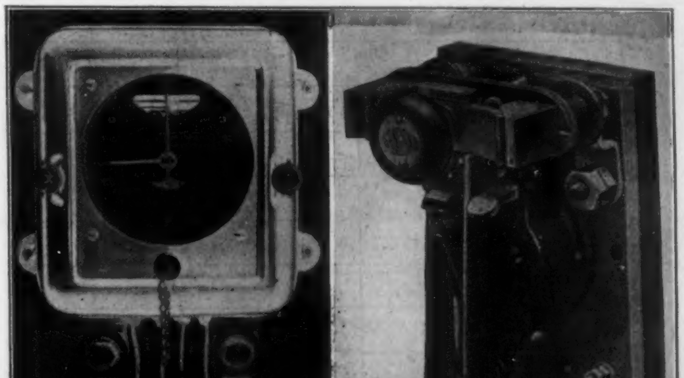


FIG. 8—AT THE LEFT AN AMPERE-HOUR METER AND AT THE RIGHT A SHUNT-TRIP CIRCUIT-BREAKER

cleaning and overhauling. This means that each meter has an overhauling every $2\frac{1}{2}$ years on the average. The Pullman Co. has several thousand meters. Their records during 1 month showed 1 failure in 800 departures, which is a fairly close approximation of the truck record.

In the early days, even when the right type of meter was used, it was the practice to install one entirely too large. Any electric meter has a tendency to run slow at low loads and this meter is no exception. It will run 2 per cent slow at 10 per cent of its rated load; below this, it falls off rapidly. Meters twice as large as should have been used were installed in many cases. This meant that the finishing rate of charge went below the range of accuracy; the meters ran slow and overcharged the batteries. The meters should not be blamed for what was simply poor engineering. We are now recommending the exact size of meter to be used in each case; we would much rather risk a burn-out than use too large a meter, and there are very few burn-outs.

Auxiliary apparatus has given trouble and this has been attributed to the meters. For example, when circuit-breakers that require too much current to release or are unreliable have been used, the zero contact mechanism has burned out and the meter blamed. A number of meters were recently reported in trouble, and in every case the cause was a broken wire in the plug cable. A solid instead of a flexible wire had been used and naturally broke after comparatively few bendings. These cases are cited to show the reason for much of the prejudice against this meter. Much of it is illogical but, on the other hand, the meter was far from perfect. The manufacturer, however, has cooperated in improving it so that today we have a different meter from the one we had even 2 or 3 years ago. For example, the meters are now dust and waterproof and it is claimed that they will operate under water. The old meters were neither dust nor waterproof and naturally gave trouble on this score. The overload capacity has been increased recently to 300 per cent of the rating, which allows the use of a much smaller meter. The aging factor or tendency to run slow has been reduced. The result is that there are fewer chances of failure with the present meter and with reasonable maintenance it will

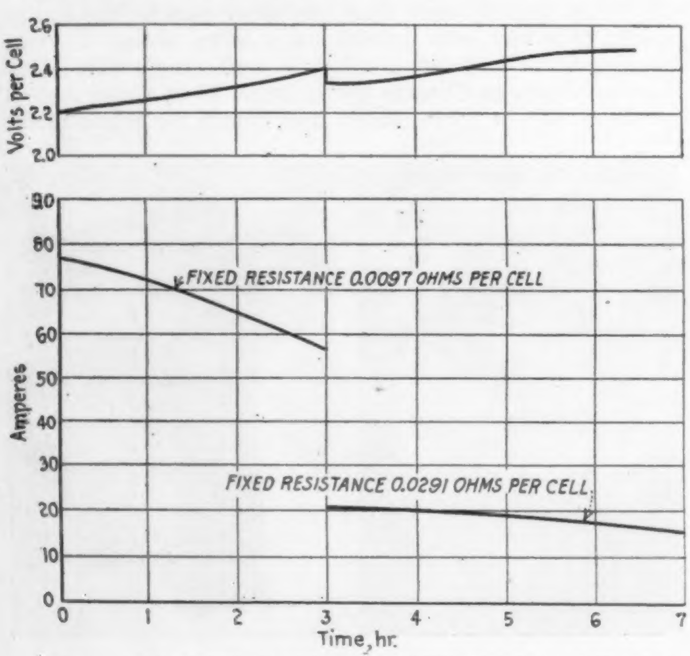


FIG. 9—CHARACTERISTIC CURVES OBTAINED DURING A TWO-STEP CHARGE OF A STORAGE BATTERY

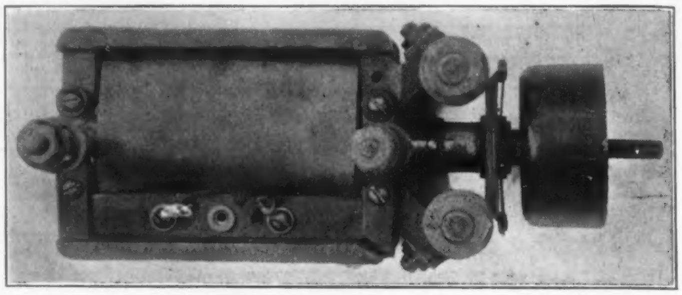


FIG. 10—A SPECIAL AUTOMATIC SWITCH THAT HAS BEEN DEVELOPED FOR USE IN CONNECTION WITH THE TWO-STEP CHARGE

give service that we have not been able to produce in any other way. It is the only satisfactory automatic storage-battery cut-off of which we know.

The view at the right of Fig. 8 shows a shunt trip circuit-breaker that has been developed for use with these ampere-hour meters. It is of the hand-closing type and simply opens the circuit when the meter reaches zero; it also has been improved considerably within the past few years.

THE TWO-STEP CHARGE

So far, in developing the automatic system, it has been assumed that a constant voltage could be selected that would produce the character of taper charge desired, but there are many cases where this cannot be controlled. Charging is often done with commercial voltages that are not what is wanted. Odd sizes of battery, such as 12, 16, 18, 24 or 30 cells are used, especially in industrial trucks, which do not fit the usual voltages. To meet this condition another system of charging which is somewhat more complicated has been developed. It is known as the two-step charge. Characteristic curves are shown in Fig. 9. In this system we start with a fixed series resistance of the proper size to give the desired current-rate and continue to a point which on an ampere-hour-input basis corresponds with the gassing point. The ampere-hour meter is equipped with a second contact which opens at this point. This opens a small self-closing circuit-breaker that shunts some part of the resistance. Additional resistance is thus cut into the circuit and the charge finished at a reduced rate. Such rate reduction can be accomplished in several ways but, where the ampere-hour meter is already installed, it might just as well be used for the rate-reducing function as well as the cut-off, so that this method has been selected for general use.

It should be noted that when the gassing point is reached the ampere-hour meter opens and does not close a circuit. This is an illustration of a characteristic that should be incorporated in any automatic system and, so far as possible, provision should be made so that if part of the apparatus fails no serious injury will result. By opening the controlling circuit instead of making it, chances of a poor contact are avoided. For example, supposing that we were to depend upon making a contact for reducing the rate, the charge would be finished at the high rate in case of a contact failure. This would overheat the battery and possibly burn it up; whereas, working the other way, if that contact is not made, it simply means that we start the charge at the low rate and, while it will require a longer time to charge, no injury to the battery will result. As another illustration, if the cut-off does not operate, nothing serious happens. The final current-rate with either system is so low that the battery could be charged for hours without causing any serious injury. We do not recommend continuing

the charge in this manner as a regular practice but, should the cut-off fail to operate occasionally, no serious injury would result.

Fig. 10 shows a special automatic switch that has been developed for the two-step charge. When the extra contact is closed, the shunt coil is energized, the circuit-breaker automatically closed and the charge starts at the high rate; or, if the meter pointer is not far enough around to close the contact, the charge starts at the finishing rate. This switch does not require manual attention but works automatically in conjunction with the ampere-hour meter. One size of breaker covers practically the entire field. It has a capacity of 100 amp., which covers nearly every case and, by shifting the connections, it can be used on either a 250 or a 125-volt circuit. Although it requires only a few hundredths of an ampere to hold it closed, the pull is so strong that it is difficult to open the switch with one hand.

While not directly pertaining to automatic charging, the ampere-hour meter has another feature that is almost as important as the charge cut-off; namely, protection against overdischarge. The dial is equipped with a red empty hand that indicates the capacity of the battery, so that the operator of the vehicle knows at any time how far the battery is discharged and what additional capacity is available. This one feature has gone a long way toward eliminating overdischarge. When a battery is overdischarged, it becomes somewhat less efficient due to the abnormal sulphation, and requires a special charge. Thus, proper charging depends upon proper discharging, and proper charging for normal conditions may be wrong charging if an overdischarge has occurred. The empty hand is normally set at about 90 per cent of the battery rating to provide some leeway and permit getting home in case the limit is reached on the road. This one feature, protection against overdischarge, is considered so vital by one mining company that the empty hand has been changed to a contact. If a motorman reaches the limit, a circuit-breaker opens and renders the locomotive inoperative. The circuit-breaker is locked and the motorman has to send for the key before he can move, so he is not apt to allow this to happen very often.

THE DISCUSSION

CHARLES H. ROTH:—How does the automobile charge-control system work out in farm lighting-battery charging, using the ampere-hour meter?

H. M. BECK:—The ampere-hour meter is probably the best device for farm lighting-plants but it is not as well suited to this service as it is to motive-power work. The farm lighting-plant loads become so low at night, when using single night-lights for instance, that the ampere-hour meter does not record the discharge. The result is that part of the load is not recorded and the battery is not really charged when the meter indicates charge. We call this getting out of step. It means that there will be a battery failure sooner or later. The ampere-hour meter has not yet been developed so that it can be thoroughly depended upon for stationary service. I have a sample meter now that I expect to test very shortly, and of which I am hopeful. It is not a mercury-type meter, but of the commutator type. It has a wider range, but unfortunately will not withstand the vibration of the motive-power service.

MR. ROTH:—Are you simply waiting for a perfect auxiliary device?

MR. BECK:—When we get a device that is suitable for the stationary plant, we will certainly push it.

G. T. BRIGGS:—How does the battery withstand tractor work, when using it for night work?

MR. BECK:—That is rather severe service. We developed a battery that will withstand it. We have installed it in a number of tractors but the field has developed slowly. The battery has a number of features. We subject the cell to a bumping test, and it actually withstands millions of bumps that will put the ordinary automobile storage-battery cell out of service in from 3 to 5 hr., as the plates cut right through the bridges and the bottom of the jar; but that tractor cell will come through in perfect shape with only a small indentation in the top of the rib. The top of the supporting bridge is made of soft rubber so that it cushions the plate. The sealing and sealing nuts are special and the jars are extra heavy. So far as I know, only two tractors in general use have adopted the battery as standard equipment.

H. L. HORNING:—We have discouraged the application of engine-starter equipment to trucks, not because we thought the storage-battery was not good enough but on account of the amount of trouble in making the engines ready to connect the generator. That attitude is simply the result of practical considerations in manufacture. In connection with motor trucks, there is no doubt that a suitable electric lighting system is becoming a necessity; the motor truck cannot fulfill its place in the transportation system without one. At present it is difficult to persuade anyone to use a battery system on tractors. There is some feeling against the use of a battery system for ignition on the motor truck, although the objection is not so strong as in the case of the tractor. I have confidence in the manufacturers of batteries and electric systems and believe that they can overcome the difficulties if they are given the opportunity. The practical considerations of operating and maintaining batteries in the country and the unfamiliarity of the public with storage-batteries have retarded their adoption. The time is coming when generator-charging outfits will be demanded.

R. B. HALL:—What is the average life of a storage-battery on industrial trucks, provided it is charged properly by the automatic system.

MR. BECK:—I referred to one battery that has been in straight industrial truck service for 3½ years and still operates well. The length of life of a battery depends not only upon the care it receives but also upon the amount of work it does. In lightly worked trucks the battery lasts much longer. Yard-locomotive work is light service. We have a battery that has been in such service 10 years. I have a battery in my own car that is more than 8 years old but the service is fairly light. We eliminate the charging problem with the automatic charging system; so the matter is largely one of how much work the truck does. The older batteries have a 1¾-in. sediment space. We now allow for about 2½-in. of sediment. The industrial-truck batteries that I mentioned are depositing sediment at the rate of about 5/16-in. per year; it is evident that the Ironclad battery has a long life. The results with the flat-plate battery are not so good. The Ironclad plate should last from two to three times the life of the flat-plate. We estimate that the flat plate will withstand 300 to 400 complete discharges and that the Ironclad plate will withstand about 1000.

MR. ROTH:—Is not 10 years a remarkable length of battery life for motive-power service?

MR. BECK:—It is better than the average.

MR. ROTH:—What is the life of batteries under the

worst and the best conditions? I understand that starting and lighting batteries are short-lived.

MR. BECK:—On the basis of the batteries renewed by one of our depots during 1 year, the average battery life was about 3 years. By automatic methods we hope to bring the average life closer to the maximum.

MR. ROTH:—A man who manufactures storage-batteries told me several years ago that the average life of a storage battery on an automobile, for starting and lighting purposes, is about 11 months.

MR. BECK:—The life of starting and lighting batteries is usually stated to be 18 months. The last statistics on this type that I have seen averaged a period of somewhat more than 2 years. The heavier type of starting battery will average from 4 to 7 years, but on account of its weight, size and cost, it is not used generally.

TALIAFERRO MILTON:—The New York statistics for all the batteries for last year show that the average battery-life, including 7 months' minimum life, was 38 and the maximum 55 months; many of them had a life of more than 40 months.

M. SCHIFF:—With reference to the industrial application of batteries for propulsion purposes, you mentioned that there might be a considerable diversity of battery sizes. How is the automatic feature carried out under such a condition?

MR. BECK:—Using the taper charge, the fixed voltage is the same no matter what the size of the battery is; it depends only on the number of cells. The series resistance varies with the size of the battery. For example, suppose we have a street truck having the proper number of cells for a 110-volt charging system and desire to arrange for such a charging system. We would install a resistance in series with the charging socket on the vehicle, figured for that particular size of battery. If the equipment is connected to a 110-volt system, the desired amount of current is obtained automatically.

MR. SCHIFF:—Do you use the constant-potential method of charging only in that particular application?

MR. BECK:—We use what we call the modified potential in that particular application.

MR. SCHIFF:—I understand that the ampere-hour meter is not used in that instance.

MR. BECK:—No, it is used only as a cut-off in that arrangement. As a means of rate control, the ampere-hour meter comes in only where we cannot control the voltage. There are many situations where we cannot get the voltage we want. In that case we use the two-step charge, which is controlled by the ampere hour meter. Where we can control the voltage, we much prefer the straight modified constant-potential system. When necessary, we can charge more quickly with the straight modified system. The best we can do safely with the two-step system is to charge in about 7 or 8 hr., but we can charge in 4 or 5 hr. with the modified system.

MR. HALL:—What is the effect of charging with the cercury-arc rectifier instead of a generator? Is it possible to use that with the automatic system?

MR. BECK:—The question of using the mercury arc is entirely one of the rectifier characteristic. The rectifier can be designed, but the minimum current limit is fixed. If the desired minimum is not too low, the rectifier affords a satisfactory charging system.

MR. SCHIFF:—In using the ampere-hour meter to control the charge, how is the charge affected at temperatures near freezing or below?

MR. BECK:—The ampere-hour efficiency is almost independent of the temperature. That is one reason this system has gained ground so rapidly. The number of watts is not independent of the temperature; that is, the charging voltage is much higher at low temperatures. If we were using a wattmeter, more watts would be needed when charging at low temperatures, but the ampere-hour efficiency is almost constant. At high temperatures the internal losses in a battery are somewhat increased. Where the amount of discharge is light, the internal losses may become an appreciable percentage, but in ordinary service we use standard fixed percentages of overcharge.

MR. SCHIFF:—What I wished to bring out is that the ampere output is actually very much lower.

MR. BECK:—An appreciable time must elapse before outside temperatures affect the commercial vehicle to any great extent. Where the battery is charged at reasonable intervals, the cells keep fairly warm. With my own car, which is kept in an unheated garage that reaches the full outside temperature in winter, I take this into consideration and allow for it. Whereas I have 120 amp-hr. available in summer, my limit is 100 amp-hr. in winter. There is not so much variation in commercial work; for instance, mine locomotives operate under fairly uniform temperature conditions and commercial electric trucks are worked hard and are out only about 8 hr. continuously in cold weather.

MR. SCHIFF:—I have noticed that temperature effect, particularly in tests of a large submarine type of battery. I have seen the capacity reduced about 30 per cent.

MR. BECK:—About 0.5 per cent per deg. at the normal rate represents the reduction in capacity. The number of ampere-hours is reduced if the temperature falls. In passenger-car service this must be taken into account. The general rule is to cut that down 25 per cent in freezing weather. The advantage of the red hand on the dial is that the driver does not need to know anything about the size of the battery or its capacity; he simply sets that red hand back. As an example of how effective this is, I cite one car that has never run out of power on the road in 8 years. That is one reason the battery is giving full capacity and has never been renewed or even had the electrolyte strengthened. This illustrates what protection against over discharge means and how the ampere-hour meter will provide such protection. The instrument is almost as important in connection with discharge as with charge. In commercial work it is felt in some large factories that the protection on discharge is more important than on charge. The Goodyear factory at Akron, Ohio, was bringing in trucks long distances for charge when the batteries were only one-quarter discharged. The company began leaving the trucks out twice as long after installing the meters, and this allowed them to work much more efficiently. This phase of the ampere-hour meter must not be overlooked; it does more than simply cut off the charge.

MR. ROTH:—If an ampere-hour meter could be made very cheaply, would it be used on the ordinary automobile also?

MR. BECK:—Yes, if it were reliable. Originally, an ampere-hour meter was used on the automobile, but it was of a cheap type and not altogether a success. However, some of those original batteries have had a wonderful length of life; some of them have lasted 7 years. We do not favor anything but the best ampere-hour meter of those available.

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

BRAKES AND SAFETY

IN a very comprehensive paper on passenger-car brakes, by J. Edward Schipper, published in THE JOURNAL for April, 1922, the general problems of brake design and the various types of brakes and brake mechanisms are discussed in a very interesting and thorough manner. This paper shows how much has been done in the line of developing brake mechanisms to meet the needs of the present-day passenger car with a reasonable amount of safety and freedom from trouble, and at a reasonable cost.

There are, however, some phases of the brake problem that seem to merit further consideration and the purpose of this article is to suggest some of them on which research of a more or less fundamental character is needed, as distinguished from the work of design and development.

The functions of a brake may be arbitrarily divided into two classes requiring somewhat different characteristics. These are

(1) Retardation of emergency stopping
(2) Absorbing power for long periods, as on grades

Much the greater amount of service of the brakes on the average passenger-car is of the first class, as in bringing the car to a stop or slowing down in traffic. The use of brakes on very short grades makes a similar demand to that made in ordinary retardation.

On some passenger cars used for touring in rough country, and on many trucks, however, the second class of service is of great importance. On long grades brakes, or the engine used as a brake, may be called on to absorb an amount of power nearly equal to the normal power output of the engine for considerable periods of time. Putting heat into a brake system by absorbing power in friction is something like dipping water into a leaky bucket. If the bucket is larger than the dipper, and we dip only occasionally, the water leaks out and does not overflow. But if we pour in a continuous stream, there is soon an overflow unless the water leaks out as fast as it flows in. This illustrates the two distinct properties of a brake system as regards its ability to take care of the heat which it receives.

First is its capacity for absorbing heat, corresponding to the capacity of the bucket. This capacity is about proportional to the total weight of the brake-drum and shoes. It does not matter how fast heat is generated if the total amount at any one brake-application is not more than the capacity of the brakes for absorbing it. The capacity is the amount of heat in British thermal units that can be absorbed without heating the drums to a dangerous temperature. The maximum amount of heat that can be developed in stopping a car at any speed is easily computed from the weight of the car and the speed. Not all of this goes into the brakes, but most of it may. However, the heat capacity of most brake systems is ample to care for this, and the real problem is that of how fast the brakes can be made to absorb the energy of the vehicle.

The requirements for the continuous absorption of power are entirely different. The heat capacity of the system is of little importance; of none, in fact, if brake application continues for many minutes. The important problem here is the heat leak, or the rate at which the brake system can dispose of the heat generated.

As the temperature of the brakes increases, the rate of heat leak increases, and the temperature will ultimately reach a point where the heat given out by radiation, convection, etc., equals the heat generated. The question then is, whether this temperature is destructive to the brakes when the power absorbed is what one could reasonably expect.

Most brakes will not meet this requirement in some parts of the Country, as Mr. Schipper points out in his paper. The area of the brake-lining and the holding power are ample for the purpose, but the provisions for disposing of heat are not. In fact, this feature seems to have been given little, if any, consideration in most brake systems.

Some of the problems that seem to need further investigation in the light of the foregoing discussion are as follows: In most cases brakes have obviously been designed like the rest of the car to meet average and not the most severe conditions. Brake failures have been responsible for many accidents and there is an insistent demand among traffic authorities in many quarters for some means of placing the responsibility for such failures. Official periodic tests of brakes have been advocated or adopted in some places. This demand has been so far justified that it is safe to assume some means will have to be found to meet it. To what extent are the designer, the production department, the service department and the user responsible for brake failures?

To what extent are accidents involving inadequate brake action dependent upon road conditions, and to what extent can the responsibility rest upon the highway designer and the driver of the car? Can traffic regulations be framed so as to place the responsibility where it belongs, and to keep the fact of this responsibility clearly before the mind? Certainly very much of the responsibility rests with the driver and fortunately he is easiest to reach. Correct answers to the foregoing questions would help in drafting regulations regarding brakes.

If there is to be a demand for brakes that will meet all usual conditions of service, there should be some accepted standard of service. Should brakes be designed so as to hold the car on any practicable grade and, if so, what should be considered a practicable grade? Is it a rational assumption that a car may be called on to coast down any grade at the same rate at which it would climb the grade? On this assumption it would be simple to fix a brake capacity in proportion to the power of the engine, the various power losses in the car being considered.

Brake-shoe pressures are rather a matter of design than of research, unless perchance there may be pressures beyond which it is not safe to go for some specific reason. But brake holding-power involves brake-shoe pressure and the coefficient of friction. The paper by S. Von Ammon in THE JOURNAL for March, 1922, describing the results of work done at the Bureau of Standards, raises one point of great importance in connection with the safety of brakes. While the normal coefficients of friction vary between wide limits, there are some conditions under which very abnormal values occur, namely when new brake-linings first become hot. In this case the coefficient may be reduced, with some lining materials, to perhaps one-fifth of the normal value. Should all brake linkages be designed to produce five times the pressure normally used, if this is possible, or should brake-linings be improved to overcome this tendency?

Much more should be known about brake-lining materials. However, this need is being met by an intensive research campaign on the part of many manufacturers of brake-

linings, which has followed the research undertaken at the Bureau of Standards.

Another question in the brake safety field is how much distance should be required to stop a car on a given road at any speed? And what is common practice in this respect? A few figures have been published showing the distance required. Some of these may be found in the Schipper paper, but the information is very meager indeed, particularly as regards different kinds and conditions of road. We believe that a systematic study of this problem for all classes of car and a wide variety of roads is of the utmost importance before any attempt is made to incorporate such figures in any form of traffic regulation that may be promulgated by any State or municipal authorities.

Other problems that may be considered in this connection are

(1) The effect of braking ability on the speed that can be maintained with safety in traffic

(2) The inherent advantages and disadvantages of four-wheel brakes

(3) The equalization of brake action on all wheels

(4) The possibilities of brake systems which will avoid skidding

Most of these topics are referred to in the following list of references.

BIBLIOGRAPHY

1. Passenger-Car Brakes. J. E. Schipper. THE JOURNAL, April, 1922, p. 273.
2. Developing a Method for Testing Brake-Linings. S. Von Ammon. THE JOURNAL, March, 1922, p. 153.
3. Conference on Brake-Lining Tests. Bureau of Standards. THE JOURNAL, July, 1921, p. 75.
4. Practice and Theory in Clutch Design. Herbert Chase. THE JOURNAL, July, 1921, p. 47.
5. Engineering Analysis of European Four-Wheel Brakes. W. F. Bradley and S. Gerster. *Automotive Industries*, June 16, 1921, p. 1267.
6. Report on Brake-Lining Conference Held at the Bureau of Standards, May 17, 1921. (Copy on file at the office of the Society.)
7. Braking Effects on Road Vehicles. F. A. S. Acres. *Automobile Engineer*, October, 1918, p. 298.
8. Retardation of the Automobile. John Younger. THE BULLETIN, January, 1917, p. 371.
9. Friction Brakes for Automobiles. R. Waring-Brown. *Automobile Engineer*, March, 1916, p. 75, and April, 1916, p. 106.
10. Jackshaft vs. Double Rear-Wheel Brakes. Arthur M. Laycock. THE BULLETIN, May, 1913, p. 216.
11. Method of Brake-Capacity Determination. S. I. Fekete. THE BULLETIN, June, 1912, p. 139.
12. Arrangement of Brakes. *Automobile Engineer*, March, 1911, p. 289.

MAY COUNCIL MEETING

THE meeting of the Council held in Indianapolis on May 8 was attended by President Bachman, Second Vice-President Young, Councilors Crane, Scott and Smith, and Past-President Beecroft.

Eighty-five applications for individual membership and 25 for student enrollment were approved. The following transfers in grade of membership were made: From Member to Service Member, Floyd B. Newell; Service Member to Member, Ernest W. Dean; Junior to Member, H. A. Schwartz, Dan R. Veazey, Walter R. Griswold; Junior to Associate, Bertram B. Webb; Associate to Member, C. W. Bassett, G. C. Brown, W. P. Hilton, Samuel Tour, Howard W. Draper, C. Eberhart, Jr., F. E. Whittelsey.

The financial statement as of March 31 showed a net balance of assets over liabilities of the Society of $124,110.89, this being $10,380.62 less than the corresponding figure on the same day of 1921. The income of the Society for the first 6 months of the current fiscal year amounted to $77,959.88. The operating expense during the same period was $92,-419.48. The expense accounts showed a net saving of $8,737.62 in comparison with the same activities for the same period of the last fiscal year.

The following appointments to the Standards Committee were made:

AERONAUTIC DIVISION
L. M. Woolson

AXLE AND WHEELS DIVISION
O. J. Rohde A. L. Putnam

BALL AND ROLLER BEARINGS DIVISION
J. T. R. Bell

CHAIN DIVISION
F. L. Morse

ELECTRIC VEHICLE DIVISION
Charles R. Skinner, Jr.

FRAMES DIVISION
E. A. DeWaters

NOMENCLATURE DIVISION
Leonard Ochtman, Jr. L. C. Voyles

NON-FERROUS METALS DIVISION
Samuel Tour

PARTS AND FITTINGS DIVISION
E. W. Weaver

PASSENGER-CAR BODY DIVISION
William Brewster

RADIATOR DIVISION
A. Ludlow Clayden

STORAGE BATTERY DIVISION
W. E. Goosling G. W. Vinal

TRANSMISSION DIVISION
C. H. Grill

The following additional subjects were assigned to the Standards Committee:

SUBJECT	DIVISION
Airplane Starting-Motor Mountings	Electrical Equipment
Battery Trays for Electric Trucks	Electric Vehicle
Cable Clips	Electrical Equipment
Carbureter Tests	Engine
Definitions	Springs
Engine Rotation	Motorboat
Gages and Gaging	Screw Threads
Magneto Flange Mountings	Electrical Equipment
Motorboat Tachometer Drive	Motorboat
Motor-Truck Cabs	Truck
Muffler Outlet-Pipe Couplings	Engine
Plain Washers	Parts and Fittings
Rubber Bushings	Electrical Equipment
Studs	Screw Threads
Stuffing-Boxes	Motorboat
Trimming Materials	Passenger-Car Body
Woodruff Keys	Parts and Fittings

J. M. Watson and C. N. Dawe were appointed to serve as representatives of the Society on the Committee on Gage Steel whose activities are conducted under the auspices of the Bureau of Standards.

Oil-Pumping

By GEORGE A. ROUND[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* DRAWINGS

OIL-PUMPING is defined and its results are mentioned. The influence of various operating conditions is brought out, particular reference being made to passenger-car service. The factors that control the rate of oil consumption are described in detail and some unusual conditions are reported. Various features of piston grooving and piston-ring design are mentioned and the effect of changes illustrated. The relative advantages of the splash and the force-feed systems as affecting the development of oil-pumping troubles are set forth and improvements suggested. A new device for reducing oil-pumping dilution troubles is described and illustrated.

OIL-PUMPING may be defined as the passing of oil into the combustion-chambers of an engine at a greater rate than it can be burned cleanly by the fuel charge. The results are spark-plug fouling and carbon deposits or in the absence of these difficulties the rate of consumption may be sufficient to cause complaints from the owners on the oil cost. The amount of oil that can be burned without trouble in any engine depends somewhat on the oil character but chiefly on the load factor and the correctness of the mixture. For example, an engine in tractor service will burn cleanly a volume of oil that in passenger-car service would cause excessive carbon deposits quickly. Again it is often the case that the engine that carbonizes in city service will burn clean in touring.

Because of the influence of the load factor it is possible to use in the more severe classes of service a type of lubricant that, while desirable from the standpoint of lubrication value and high economy, would cause undesirable carbon deposits in engines operating under more moderate conditions. In cars having a high acceleration rate the engine load factor under normal conditions has become very small. With the rich mixtures commonly used conditions are most unfavorable for burning cleanly any oil passing the pistons. This type of engine is also prone to knock readily with slight carbon deposits due to the high compressions usually employed. Consequently, it has been necessary to reduce the oil consumption in such engines to a point that is undesirably low from the dilution standpoint.

To assure good lubrication and to offset the effects of dilution it is desirable that the crankcase oil be renewed at reasonably frequent intervals. This can be accomplished in two ways; by (*a*) periodic draining of the entire supply or (*b*) the frequent addition of fresh oil to replace that used. In cars that show a very low oil-consumption rate the first method should apply, but in spite of repeated instructions to the owner to drain the old oil frequently, he finds it such a disagreeable task that it is not done as often as it should be, and contamination of the oil to an excessive degree is the result. It is a fact, however, that in many engines the rate of oil consumption increases to an undesirable extent after a comparatively short period of service, and it is in the control of this that we are chiefly interested.

FACTORS CONTROLLING OIL-PUMPING

The amount of oil passing an engine piston in a given time depends upon the following principal factors: (*a*) the amount of oil thrown to the cylinders, (*b*) number of piston strokes, (*c*) the efficiency of the means for piston drainage, (*d*) the ring fit, (*e*) the oil viscosity and character and (*f*) the vacuum in the cylinder. In his paper presented at the Annual Meeting last January,[2] Mr. Ricardo stated that he had been unable to find that the vacuum in the cylinder had any effect on oil-pumping. There is, however, some evidence to the contrary.

A few years ago an engineer working out a device to prevent oil-pumping sealed the crankcase of an engine against air leakage and maintained a vacuum in it equal to that in the intake-manifold. The tests conducted showed that the oil consumption could be cut in half when the vacuum was maintained. Smoking when accelerating after prolonged idling was eliminated, showing that the amount of oil passing the pistons was negligible. While that method of reducing consumption was impracticable, it showed that the vacuum in the cylinder was a factor affecting the amount of oil passing the rings, particularly in worn engines, and the idea has been applied successfully in another way that will be described later.

Experience seems to indicate that it is not possible to establish a definite rule for the variation in oil consumption with changes in the viscosity and the character of the oil used. In general the consumption decreases with an increase in the viscosity but in several tests we have encountered the reverse condition, under much the same conditions of design. In the engines in which this occurred the pistons were of cast-iron with no grooves or relief on the piston skirt. The ring next above the piston-pin was fitted somewhat loosely in its groove in the bottom of which was drilled a series of return holes. In one engine using force-feed lubrication and one employing splash the lower-viscosity oils invariably showed the best economy. The consumption figures for the first case, a small six-cylinder engine, were 1.37 lb. in 10 hr. for the lighter oil, and 2.1 lb. for the heavier under similar operating conditions. For the second engine, a small water-cooled single-cylinder lighting unit, the consumption of lighter oil was 0.5 lb. in 10 hr. and the heavier 2.6 lb. The consumption figures in both cases were corrected to allow for dilution. The viscosities at 210 deg. fahr. were 45 sec. Saybolt for the lighter oil, and 58 sec. for the heavier. Apparently the resistance of the lubricant in passing behind the ring to reach the return holes is the controlling factor, as with a different form of oil return the behavior of the oils is entirely normal. The character of the oil as affecting its evaporation rate, particularly at high temperatures as in tractor or heavy-duty truck service also has a bearing on the consumption. However, the more heat-resisting oils are not always desirable because from that very characteristic they may

[1] M.S.A.E.—Assistant chief of the engineering division of the automotive department, Vacuum Oil Co., New York City.

[2] See THE JOURNAL, May, 1922, p. 321.

not burn cleanly under moderate load conditions. In the majority of cases a compromise is desirable and that must be worked out largely on the basis of practical experience.

The Importance of Piston-Ring Fit

Proper piston-ring fit is one of the most important factors controlling oil consumption. When the rings fit correctly, the matters of clearance and provision for drainage become of minor importance. The chief points in regard to ring fit are the amount and uniformity of the tension of the ring and its clearance in the ring-groove. Experience has indicated that ring end-clearance is of minor importance in controlling oil consumption. If the rings are fitted with a minimum gap when new, the increase due to wear will have a negligible effect as compared with the other changes that take place.

Other factors being equal, the amount of oil consumed can be varied through a considerable range by changes in the ring pressure. This was clearly shown by some experiments on aircraft engines that were using too much oil. After fitting the bearings as closely as possible and cutting the pressures to a minimum, it was found that a 30-per cent reduction in the oil consumption could be effected by reducing the contact area of the rings about one-third as shown at the left of Fig. 1. It is interesting to note that this method was equally as effective as the other methods of reducing the area shown in the two central views of this illustration. This is contrary to the common opinion but it was checked several times in different engines.

The amount of pressure is, however, fairly definitely limited by the rapid wear that takes place when the ring tension becomes sufficient to cause all the oil to be scraped off. The pressure to cause this varies with the character of the lubricant and the nature of the cylinder and ring metals. A reasonable working limit, however, in terms of the pressure required to close the ring is approximately 2 lb. per in. of diameter for a ring of 1/4-in. width and the other sizes are in proportion. Uniformity of tension throughout the ring is of maximum importance and many cases of oil-pumping have been traced to the lack of this, due to either defective rings or their being distorted during assembly.

From recent experience, it is apparent that in some cases at least, not nearly enough attention is being given to ring inspection or to care in assembling them on the pistons. For instance, in one group of engines taken down to determine the cause of oil-pumping, a large number of rings were found bearing in spots only and of the new rings in stock, over 25 per cent proved defective. A little more care given to this detail would be of great benefit.

As to the relative merits of narrow and wide rings in controlling consumption, for equal wall-pressures there seems to be no difference. Because of their lighter weight the narrow rings do not wear the ring-grooves as rapidly, a decided advantage in aluminum pistons and one that gives them preference from that standpoint. In cast-iron pistons, a wide ring gives excellent results once it has worn to a fit.

The fit of the rings in their grooves is of prime importance. Our experience indicates that in engines where the amount of oil thrown to the pistons is constant throughout the engine life, as with splash systems, the oil consumption increases directly with the increase in the ring-groove clearance. Therefore, unless the lubricating system and the pistons are designed so that a minimum quantity of oil reaches the piston-rings throughout the normal life of the engine, oil-pumping will increase as the rings wear loose in their grooves.

With cast-iron pistons and rings of moderate width, the wear is not particularly rapid, although in motor-truck work it becomes noticeable in less than a year of service as a general rule. With aluminum pistons the wear is much more rapid and unless the oil supply is controlled effectively by other means, excessive consumption results. To avoid this increase in ring-groove clearance, rings have been made that, due to their design, stay tight in the grooves throughout their life. A typical ring of this kind is shown in Fig. 1 at the right. This ring we have found particularly effective in overcoming oil-pumping under a wide range of conditions.

The benefits of such rings are usually slight when compared with new and tightly fitted rings but show up to a marked degree in long service. One example of this was a 3 1/2 x 5-in., four-cylinder engine that was fitted with rather heavy aluminum pistons having approximately 0.007-in. clearance. Using two plain rings above the piston-pin and one scraper ring, the oil consumption was about 60 miles per qt. of a medium-bodied oil. When the second ring was replaced with a ring of the type mentioned, the distance increased to approximately 150 miles per qt. and remained at that point until the pistons were discarded because of excessive slap after some 8000 miles of service. During this time the clearance of the other rings had increased to a marked degree. Numerous other similar cases are on record. While the scraper or skirt ring apparently has only a slight effect in most cases in reducing the oil consumption, it is of value in keeping the wear of the cylinder bore uniform.

Oil Grooving

The use of a properly designed drain groove with adequate return-holes is a great help in keeping oil consumption low, particularly after the rings have become somewhat loose in their grooves. While it has sometimes been the case with new engines that oil-return grooves have not shown much saving, in engines that have become somewhat worn they have proved helpful almost invariably.

Several different forms of return groove have been used. One of the earliest and most common types is shown in Fig. 2 at the left. For this the bottom edge of the lower ring-groove is beveled off slightly and a series of holes drilled from this bevel into the inside of the piston. Where the drain holes are of sufficient size and number and the space for oil collection provided by the bevel is of adequate size, this form of return is effective. It has, however, the decided disadvantage of reducing the area that supports the ring. Consequently, the ring-groove clearance tends to increase more rapidly than it otherwise would, thus defeating the purpose of the groove.

A more desirable form of groove is shown in the center of Fig. 2. With this, a full support of the ring is provided, together with a larger space for the accumulation of the excess oil. In some engines the most effective return has been by holes drilled in the bottom of the lower ring-groove as indicated at the right in Fig. 2. In these the best results have been secured when the ring was slightly loose in its groove and when using a light-bodied oil. Examples of this have already been given.

The size, number and location of the drain holes is an important factor. Holes smaller than 3/32-in. diameter are apparently ineffective unless a large number are used. With 3/32 or 1/8-in. holes, spaced approximately 1 1/2 in. apart, satisfactory results are generally obtained.

With certain V-type engines, it has been found that the drain holes are effective only when placed as indicated in Fig. 3. When the holes are drilled on 360 deg. of the groove, the consumption increases. This is an unusual condition and one that has not yet been explained fully but is now under development.

The foregoing comments regarding grooving apply to the more conventional types of cast-iron and aluminum pistons. The latest forms of constant-clearance aluminum pistons are inherently self-draining and should perform satisfactorily provided the rings do not wear loose in their grooves too rapidly and, what is still more important, that the amount of oil thrown to the piston is not excessive.

Lubricating Systems

During the past few years we have seen a gradual trend toward the use of the force-feed lubricating system in which the connecting-rod dip is eliminated. The advent of the V-type engine made the use of this system necessary to assure a uniform distribution to the cylinders. Many have followed this lead in adopting pressure systems for other types, but the results as regards oil-

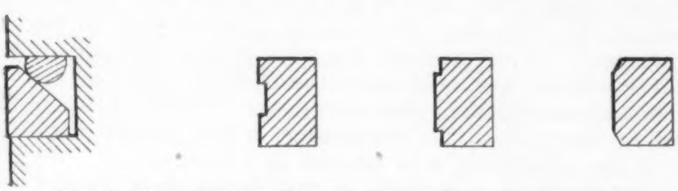


Fig. 1—Examples of Typical Piston-Ring Construction
The First Three Views Beginning at the Left Show How the Pressure of the Piston-Rings Against the Cylinder Wall Is Increased by Reducing the Contact Area, While the Illustration at the Extreme Right Shows a Ring That Is Designed to Remain Tight in the Groove

pumping have not always been as happy as desired. The great disadvantage of the force-feed system from an oil-pumping standpoint is that the amount of oil thrown to the cylinders increases in proportion to the wear of the bearings and that unless special provision is made to offset it, the amount of oil supplied at light loads is far in excess of the requirements. In contrast to this the oil throw with splash systems remains constant throughout the engine life and may be proportioned so that, while the bearings receive ample lubrication, the cylinders are not over-lubricated beyond the ability of the rings to hold consumption to a reasonable figure, even when worn.

Perfection of bearing fits has done much to reduce the oil bleed in force-feed systems, but this is often nullified later by poor repair work and by excessive pressures. In connection with connecting-rod bearing leakage the ideal design uses no shims. Where these are employed either the lead-edged or plain brass shim is satisfactory only when fitted properly. If a bearing fails from lack of oil the lead edge will be destroyed and unless the bearing is refitted by a repairman familiar with the design that fact may be overlooked with resultant trouble. For truck and tractor service this shim seems less satisfactory.

In the conventional force-feed system employing a spring-controlled bypass, the capacity of the pump is far in excess of the bearing requirements. Consequently, this excess must be taken care of by the relief-valve, the adjustment of which determines the pressure in the system.

When the car or truck is delivered to the user, the pressure is set to give good results with tight bearings. Once he has the normal pressure as shown by the gage

Fig. 2—Some Forms of Oil-Return Grooves

fixed in his mind, it is extremely difficult to convince either the user or the average repair-man that the drop in pressure that occurs as the bearing clearances increase, is not a danger signal. It is still more difficult to convince either of them that the proper remedy for the oil-pumping troubles that accompany the pressure drop can be overcome in part by a further reduction in the pressure, without endangering the engine.

In some cases, fixed adjustment relief valves are used and no pressure gages are employed. With such designs wide variations in the actual pressure maintained are certain to occur unless the relief valves are carefully calibrated, after assembly. In one case recently investigated, the opening pressure varied from 2 to 25 oz. in 10 valves taken at random from engines and stock. A test showed that the 2-oz. valves provided ample lubrication while with the higher tensions over-oiling was common. More rigid inspection of this important detail would eliminate a source of much trouble and expense to owners and dealers.

The effectiveness of a reduction in the oil pressure in worn engines is very marked, as the following cases show. In the first an eight-cylinder car, carrying normally about 12-lb. pressure under all conditions of speed and temperature, used about 1 qt. of oil for every 50 miles run and was carbonizing badly. Dropping the pressure to 5 lb. doubled the oil mileage and practically eliminated the carbon trouble.

The other case was a fleet of motor trucks that were giving excessive carbon trouble. The pressure in these engines ranged from 5 lb. when idling to 8 or 9 lb. at governed speed. When the bypass-valves were set to maintain an idling pressure of 1 lb. and a maximum of from 4 to 5 lb., the carbon trouble was eliminated, the average mileage per quart increased from 18.5 to 31.6, and no bearing trouble developed.

The experiences we have had with pressure systems indicate that a far smaller amount of oil is required to

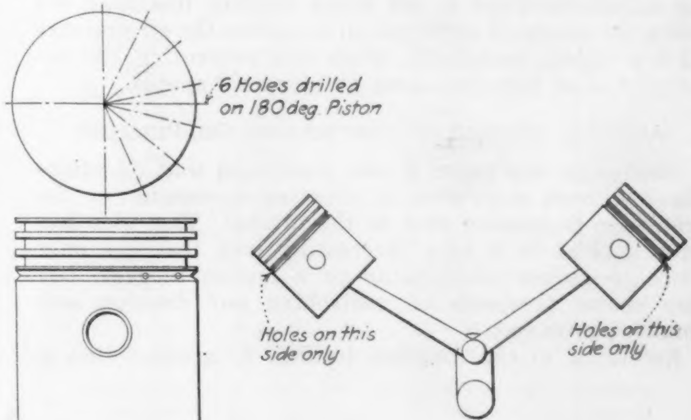


Fig. 3—Drawing Showing the Location of Oil Holes on a V-Type Engine

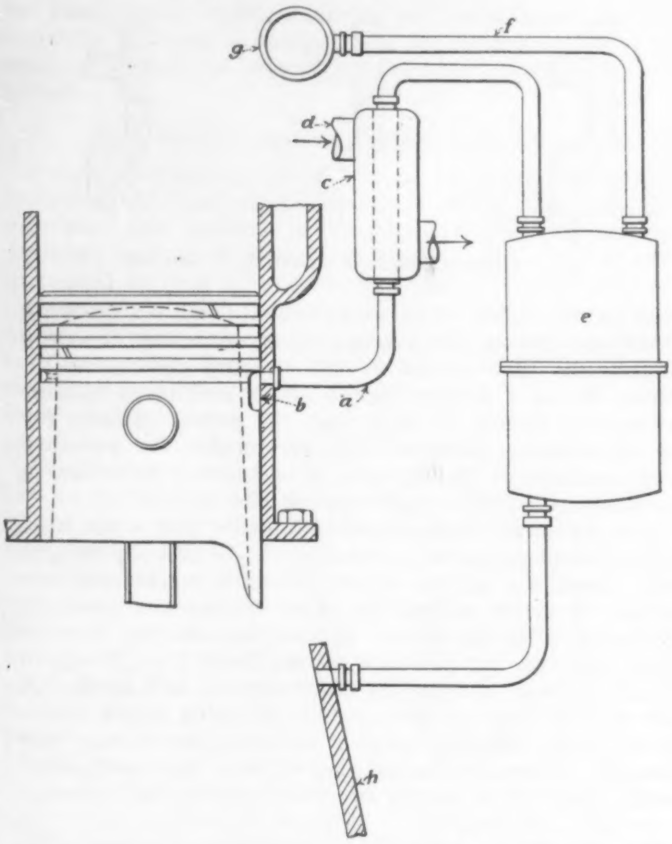


FIG. 4—DIAGRAMMATIC SKETCH OF A DEVICE DEVELOPED BY THE AUTHOR FOR OVERCOMING OIL-PUMPING AND DILUTION

lubricate a bearing adequately, and incidentally the cylinders and the pistons, than is generally supposed. Consequently, a marked reduction in the pressure and the volume of the oil supply will be of material advantage in reducing the oil consumption.

It may be contended that to meet the demands of extreme loads and speeds a large volume of oil is needed. There is some question in regard to this, as in many instances we have subjected engines to just such tests with a reduced oil supply, and always without damage. However, if it is felt that this over-supply is needed, some form of throttle or vacuum control of the oil pressure should be used to take care of the prevailing periods of light load.

Another method, which is more simple, is to provide an adjustable bleed in the pump delivery line that will allow the escape of sufficient oil to reduce the oil pressure at low speeds materially, while not preventing the development of high pressures at increased speeds.

A Novel Method of Controlling Oil-Pumping

Earlier in this paper it was mentioned that oil-pumping had been controlled by creating a vacuum in the crankcase to balance that in the intake. This idea has been applied in a very interesting way that not only tends to reduce oil-pumping to a marked degree, but also shows prospects of controlling our dilution and emulsion problems.

Referring to the diagram in Fig. 4, a connection *a* is made to the cylinder wall at a point that coincides with the lower limit of the travel of a groove *b*, located on the piston below the ring next above the piston-pin. A short vertical groove intersecting the circumferential groove and in line with the hole for the connection, is also cut in the piston. The connection *a* leads to a heater *c* that is connected to the exhaust manifold by the pipe *d* and thence to a separator *e*. From the upper part of this device a connection *f* is made to the intake-manifold *g*. The lower part of the separator is cut off from the upper by a valve and leads to the crankcase *h*. On the side of the piston opposite the connection a hole is drilled through the groove into the inside of the piston.

In the operation of the device the suction in the manifold is communicated to the groove on the piston, while the latter is near the bottom of its travel, and during approximately 90 deg. of crank rotation. The vacuum created draws into the heater any oil or oil and fuel mixture that collects in the groove and with it a small amount of air. In passing through the heater the mixture is heated by the exhaust to a temperature of about 375 deg. fahr. On reaching the separator any fuel or water present in the oil is evaporated and carried into the manifold. The remaining oil is passed into the bottom of the separator from which it flows back to the crankcase. To maintain on the other pistons the vacuum that would be destroyed when the upward travel of the piston uncovered the connection to the heater, a lip is cast or fastened on the lower edge of the piston. This keeps the connecting hole covered at all times.

The results of some tests of this device on a number of different cars and trucks of varying lengths of service and conditions, show an increase in mileage per gallon of oil ranging from 100 to 400 per cent. These tests were not short runs, but were carried on over periods of a year or more. As the effect of this device in connection with dilution is to be presented in another paper at a later date, it will only be mentioned in passing. However, the results seem to be very satisfactory and show that we have in prospect another method of solving the dilution problem.

Conclusions

(1) The lubrication requirements of engines, particularly in passenger cars, do not demand the volume of oil supplied by force-feed systems. Under normal operating conditions the volume of fuel burned is inadequate to consume completely the amount of oil passing the pistons

(2) Oil-return grooving is desirable in all cases and special rings may be required to control the excess oil supply caused by the wear in uncontrolled force-feed systems

(3) More rigid inspection of piston-rings and greater care in fitting them on the pistons will remove a common cause of oil-pumping. Oil-pressure relief valves should be carefully calibrated and set for lower pressures

(4) Some form of throttle or vacuum control is essential with force-feed systems, particularly when used on passenger-car engines

(5) Oil-pumping and dilution can be reduced by a new device that draws from the pistons any excess oil or liquid fuel present, the latter being driven off by heat and delivered to the intake-manifold while the oil is returned to the crankcase

Oil Consumption

By A. A. Bull[1]

Semi-Annual Meeting Paper *Illustrated with* Charts and Photographs

The object of the paper is to consider some of the fundamental factors that affect oil consumption; it does not dwell upon the differences between lubricating systems. Beyond the fact that different oils apparently affect the oil consumption and that there is a definite relation between viscosity and oil consumption, the effect of the physical characteristics, or the quality of the oil, does not receive particular attention.

The methods of testing are described and the subject is divided into (*a*) the controlling influence of the pistons, rings and cylinders; (*b*) the controlling influence of the source from which the oil is delivered to the cylinder wall. The subject is treated under headings that include the piston-ring; the effects of oil-return holes, side-clearance and ring motion; thin rings; influence of piston fit; efficiency of the scraper-ring; ring and cylinder contact; carbonization and spark-plug fouling; oil-supply control; influence of oil viscosity; effects of dilution; external oil leaks and breather discharge, and influence of controlling lubrication in proportion to throttle opening.

The subject of oil consumption, relating particularly to automotive engines, is one of importance. I realize the attention and effort that have been directed toward a solution of many of its problems by both the oil companies and the engine and vehicle builders.

From the owner's standpoint, the question of oil consumption in all its phases becomes of interest sooner or later. He is perhaps more interested in the troubles that he has experienced that result from excessive oil consumption or, as it is generally understood, oil-pumping. Over-oiling of this character, which occurs under average driving conditions and causes excessive carbon deposit and fouling of the spark-plugs, results in unsatisfactory operation of the engine. However, these symptoms are not necessarily characteristic, and engines that are not directly subjected to these troubles will use excessive quantities of oil; from an economic standpoint, this phase warrants consideration. The economic viewpoint of oil consumption is more important in truck and tractor engines, or those that are used in commercial service, where oil consumption affects operating costs. The object of this paper is to consider some of the fundamental factors that affect oil consumption and it will not dwell extensively on the difference in lubricating systems which, as will be self-evident, have an influence on oil consumption. Different oils apparently affect oil consumption and there is a definite relation between viscosity and oil consumption which can be attributed to several causes that will be discussed but, beyond this, the effect of physical characteristics or quality of the oil will not receive particular attention.

The extent of dilution has an important bearing on oil consumption in its effect in lowering the viscosity of the oil so that it cannot be controlled as readily. In passenger-car engines particularly the operator is likely to believe that his oil consumption is very small for the reason that the added diluent compensates for the actual oil consumed.

Apart from over-oiling as evidenced by excessive carbon, fouled spark-plugs and the like, what constitutes reasonable oil consumption? In commercial service, where operating costs must be kept to a minimum, perhaps the oil consumption should bear a definite relation to fuel consumption. The actual amount of oil required to lubricate the engine properly, particularly the pistons and cylinders, is surprisingly small and, because the oil pumped to the combustion-chamber is the chief source of oil consumption, attention necessarily is directed to controlling the extent to which oil reaches the combustion-chamber. The wide variation in oil consumption with different heavy-duty engines is well represented by the figures in Table 1. These figures have been collected from engines of different types, working, in most cases, under actual service conditions. The basis of comparison is the oil consumed per brake horsepower hour and, while this is not a direct indication of the relative con-

TABLE 1—OIL CONSUMPTION

Engine	Type	Speed, r.p.m.	Horsepower	Hours Operated	Oil Consumption, gal. Total	Per Hr.	Per B. Hp-Hr.
A	Tractor	....	....	97.00	......	0.2070	......
A	Tractor	....	....	379.57	......	0.1430	......
A	Tractor	....	....	426.59	......	0.1370	......
A	Tractor	....	....	2,997.00	......	0.1500	......
S	Tractor	....	20.0	37.00	9.7500	0.2630	0.01320
F	Tractor	....	18.0	34.00	3.7500	0.1100	0.00620
WB	Tractor	....	25.0	44.00	3.3800	0.0770	0.00370
C	Tractor	....	18.0	40.00	8.0000	0.2000	0.01110
C	Tractor	....	27.0	37.00	11.5000	0.3100	0.01150
AR	Tractor	....	30.0	35.00	15.5000	0.4420	0.01470
AR	Tractor	....	20.0	44.00	5.0000	0.1120	0.00560
RI	Tractor	....	16.0	33.00	4.3800	0.1330	0.00830
I	Tractor	....	20.0	30.00	8.0000	0.2670	0.01330
I	Tractor	....	16.0	32.00	4.7500	0.1490	0.00930
N_1	Truck[a]	....	37.0	5.00	0.4250	0.0850	0.00230
N_2	Truck[a]	....	30.0	4.00	0.0935	0.0234	0.00580
N_3	Truck[a]	....	27.5	4.00	......	......	......
A	Truck	1,500	33.0	1,000.00[b]	......	0.1200	0.00364
A	Truck	1,200	35.0	500.00[b]	......	0.0500	0.00143

[a] Maximum dilution, 1.5 to 2.0 per cent.
[b] Continuous.

[1] M.S.A.E.—Chief engineer, Northway Motor & Mfg. Co., Detroit.

sumption, it is based on the assumption that the engine speeds are nearly the same and that the horsepower output therefore indicates the size of the engine.

Oil consumption in passenger cars usually is computed on the basis of miles per gallon. It is possible to obtain figures as low as 50 and as high as 2000 miles per gal. As indicated previously, in ordinary service with low average speeds, the oil consumed may be negligible; in fact, it is possible that more fluid can be taken from the crankcase after service than was originally put in. On the other hand, engines or cars that will exhibit this characteristic will consume unreasonable quantities at continued speeds of 30 m.p.h. or over. An instance of this character which indicates a fairly general condition is indicated in Table 2.

TABLE 2—OIL CONSUMPTION

Car No.	1	2	1[c]	2[c]
Number of Cylinders	8	8	8	8
Average Speed, m.p.h.	15	15	40	40
Total Miles Run	600	500	50	50
Oil Consumption[d]	0	0.5	4-5.75	11
Dilution per Gallon, per cent	40	10	0	0
Oil Consumption, Excluding Diluent, miles per gal.	860	835	92	581

[c] Same cars on a speedway, at 40 m.p.h.
[d] The consumption for Cars Nos. 1 and 2 is given in gallons, in pounds and ounces for Car No. 3 and in ounces for Car No. 4.

METHODS OF TESTING

In the study of oil consumption there are several methods in which comparative tests can be conducted. First, of course, tests under actual operating conditions are necessary to determine the average oil consumption and the general efficiency of the lubricating system. However, this method is not satisfactory in studying fundamentals, the chief difficulty being that it is not possible to have or control similar conditions of operation and equipment. Several methods were used in making observations in the laboratory.

(1) Obtaining the oil consumption under working conditions, properly controlling all the affecting factors
(2) Measuring the quantity of oil pumped through the exhaust of individual cylinders
(3) Studying the extent of the oil passing the pistons to the combustion-chamber with the cylinder-head removed

Method (3) gives very good results when checking

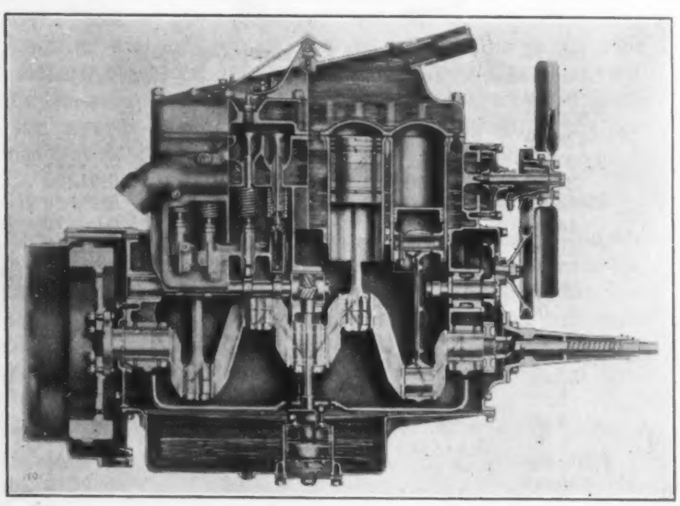


FIG. 1—SECTIONAL VIEW SHOWING THE CONSTRUCTION OF THE ENGINE UPON WHICH THE TESTS WERE MADE

fundamentals and is related closely to the results obtained by methods (1) and (2). In any case, in conducting oil-consumption tests it will be found that considerable variation in results is likely to be obtained unless great care is taken and until some fundamental has really been found. For this reason any small improvements or changes in oil economy that may be effected by the use of or changes in detail are not to be relied upon.

In studying the extent of oil-pumping by an observation of the amount collecting on top of the piston, the quantity of oil passed by is in excess of that which is used under working conditions, which assumption is drawn from the relative quantities of oil used. Every stroke of the piston functions the same, not being affected by the pressures existing in the cylinder during the cycle, the compression, expansion and exhaust pressures evidently having a beneficial influence and exercising more control than the effects of suction.

Before proceeding to a complete analysis, it is necessary to separate the subject into two classes (*a*) the controlling influence of the pistons, rings and cylinders; and (*b*) the controlling influence of the source from which the oil is delivered to the cylinder wall.

It generally is assumed that the piston-rings exercise the most important part in controlling the extent of oil-pumping. This is substantiated by the extraordinarily large number of different types of ring that claim in one way or another to control the oil consumption and eliminate the troubles usually associated with over-oiling. There are well established definite and fundamental principles that must be observed in the application of the rings to the piston, and the problem is one of maintaining the efficiency indefinitely. Unfortunately, the majority of replacement rings have no fundamental quality in themselves that affects or exercises control over the oil consumption and, in most instances where replacements are made, equally good results would be obtained by employing ordinary plain piston-rings.

An engine upon which some of the tests were conducted is illustrated in Fig. 1; its construction is self-evident. It has a full pressure-feed lubricating system, separate cylinder sleeves and the oil reservoir is divided into several compartments to filter the oil adequately.

THE PISTON-RING

While the piston-ring primarily is used to retain compression in the cylinder, it has become necessary that the ring regulate or control the extent of the lubrication of the cylinder walls. Let us consider for a moment the action of the piston-ring in traveling the surface of the cylinder already coated with oil-film. During the expansion and suction strokes, when the ring is traveling down, it is presumed that it scrapes or pushes the excess oil adhering to the cylinder walls in front of it. In any case, this is exactly what the ring is required to do. The factors that apparently affect the efficiency of the ring in performing this duty are the thickness of the oil-film and the volume of oil, the unit pressure between the ring and cylinder, which determines its ability to break-down the oil-film, and the character of the leading edge of the ring.

The thickness of the oil-film depends upon the viscosity. In this connection it should be emphasized that the temperature of the oil in the main body of the oil-pan is not necessarily at the same temperature as the oil-film on the cylinder wall and, except under conditions where the cylinders are relatively cool, there is little variation in the viscosity on the cylinder wall. Quantitatively, of course, there will be more or less oil on the cylinder wall, depending upon the extent to which the oil is thrown from the connecting-rod and crankshaft bearing which,

Fig. 2—Test Results Obtained at a Speed of 500 R.P.M., Oil and Water Temperatures of 109 and 127 Deg. Fahr. Respectively and an Oil Pressure of 12 Lb.
Cylinders Nos. 1 and 2 Were Equipped with a Single Piston-Ring and Had No Return Holes; the Other Two Cylinders Had Standard Three-Ring Equipment

as succeeding analysis shows, exercises a very decided influence.

The unit pressure between the ring and the cylinder wall has some effect on the efficiency of the ring in displacing the oil from the cylinder wall. With conventional cast-iron piston-rings, the wall pressure necessarily is limited by the stress that can be imposed on the ring material if it is to operate against the cylinder wall due to its inherent elasticity. The tension that can be obtained is adequate, although the increase in the unit pressure, obtained by a relief or a groove in the face of the ring or by the use of thin rings with a supplementary spring, will improve the scraping efficiency. As succeeding tests will indicate, however, this feature in itself is not sufficient to give the proper results. The character of the leading edge of the ring also has its effect and, unless it is relatively sharp, the ring will ride over the surface of the oil-film. For this reason a ring with a bevel is of benefit and, on the up-stroke, the ring will ride readily over the film. Along these lines, it is believed that the type of ring having an inherent twist, so that the unit pressure on the bottom edge is in excess of that on the top, may be beneficial.

Effect of Oil-Return Holes

Substantial force is required to displace the oil-film adjacent to the edge of the ring, and it is evident that a high pressure exists which, unless suitably relieved by the presence of oil-return holes, breaks-down the seal be-

Fig. 3—Test Results Obtained with the Same Equipment as in Fig. 2 at a Speed of 1000 R.P.M.
The Difference in the Oil Deposits in Cylinders Nos. 2 and 3 as Compared with Fig. 2 Should Be Noticed

tween the face of the ring and cylinder. Reference to Figs. 2, 3 and 4 clearly illustrates the effect of oil-return holes. Fig. 5 shows the same ring equipment operating at the normal oil temperature of 142 deg. fahr., but with oil-return groove and holes immediately below the ring. The improvement is evident. The ring used in the preceding test had a good fit in the groove. However, it is recognized that sidewise clearance increases with use.

The diagrams and photographs that refer to the test results are presented as representative selections of a considerable number of tests. Both photographs and scale diagrams are shown in some instances. In some cases, scale diagrams alone are shown because they offer a better comparison. The variation in ring construction is made on cylinders Nos. 1 and 2, the same rings being used in each case and being placed in the same circumferential position. Cylinders Nos. 3 and 4 have standard three-ring equipment, except where otherwise noted. Cylinders Nos. 1 and 3 have similar oil distribution from the crankshaft, while cylinders Nos. 2 and 4 differ as will be explained later. The relative effect of change in ring construction on the amount of oil-pumping with the different oil distribution can be compared in cylinders Nos. 1 and 2.

In Fig. 2 cylinders Nos. 1 and 2 were equipped with one ring and had no oil-return holes. Cylinders Nos. 3 and 4 were provided with standard three-ring equipment. The oil pressure was 12 lb. per sq. in.; the oil temperature, 109 deg. fahr.; the water temperature, 127 deg.

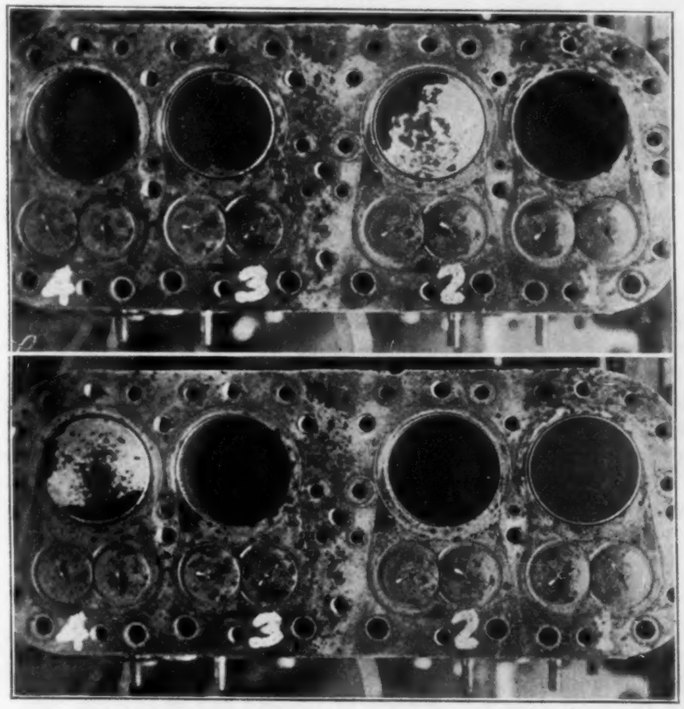


Fig. 4—Photographs of the Oil Deposits Shown (Above) in Fig. 2 and (Below) in Fig. 3

fahr.; the speed, 500 r.p.m.; and the duration of the test, 3 min.

In Fig. 3 the same conditions prevailed as in Fig. 2, except that the speed was 1000 r.p.m. and the duration of the test was 1 min. The difference in the increase in the oil deposit on cylinders Nos. 2 and 3 should be noted.

In the upper portion of Fig. 5 the oil pressure was 12 lb. per sq. in.; the oil temperature, 156 deg. fahr.; the

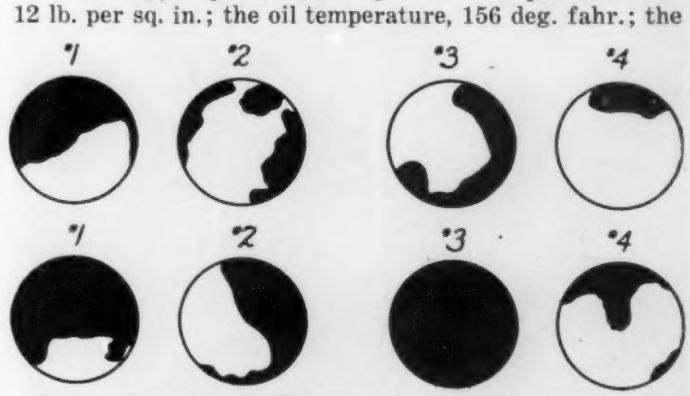


Fig. 5—Test Results Obtained (Above) at 500 R.P.M. and (Below) at 1000 R.P.M.
Cylinder No. 1 Had One Ring with Oil Holes Immediately below the Ring

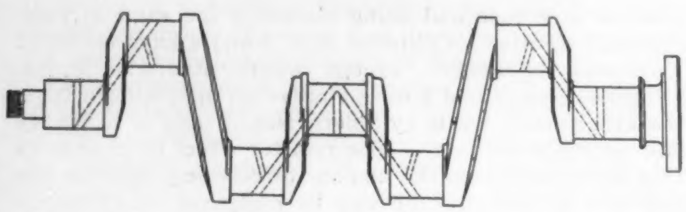


Fig. 6—Crankshaft in Which Oil-Hole Location Could Be Changed for the Experiments

speed, 500 r.p.m.; and the duration of the test, 3 min. Cylinder No. 1 had one ring, with oil-return holes immediately below the ring. In the lower portion of Fig. 5 the same conditions prevailed as in the upper portion, except that the speed was 1000 r.p.m. and the duration of the test was 1 min.

Fig. 6 shows the crankshaft, in which the oil-hole location could be changed during the experiments.

Fig. 7 illustrates part of the equipment used for measuring the quantity of oil pumped through the exhaust.

Effects of Side-Clearance and Ring Motion

Digressing for a moment, let us consider the effects of side-clearance on the ring. During the down-stroke of the piston, the ring is contacting with the upper face of the groove leaving the clearance at the lower edge; on the up-stroke, the condition is reversed. Having recognized that oil pressure is built up on the under side of the ring, it follows that, if the ring is not seating on the groove, the oil will pass through to the rear of the ring.

The three factors which control the movement of the ring relative to the piston are the

(1) Inertia of the ring itself or its resistance to acceleration
(2) Friction of the ring against the cylinder wall
(3) Effects of pressure within the combustion chamber

Fig. 8 shows the inertia and friction forces operating on the ring at 1000 r.p.m., indicating that the position of the ring does not change in respect to the groove. At higher speeds, however, the position changes before the end of the stroke.

The effect of side-clearance on oil-pumping is illustrated in the two portions of Fig. 9. The oil is pumped up very rapidly, with or without an oil-return hole below the ring. At 500 r.p.m. piston No. 1 is covered completely in 1 min., but it becomes full in 0.3 min. at a speed of 1000 r.p.m. In the upper portion of Fig. 9 cylinders Nos. 1 and 2 each have one ring with a 0.005-in. side-clearance and cylinders Nos. 3 and 4 are provided with standard three-ring equipment. At a speed of 500 r.p.m. and a duration of test of 3 min., cylinder No. 1 fills up in 1 min. In the lower portion of Fig. 9 the conditions are the same as in the upper portion, except that the speed is 1000 r.p.m. and the duration of test 1 min. Cylinder No. 1 fills up in 0.3 min. Fig. 9 should be compared with Fig. 5.

In this connection consideration should be given to the extent of the side-clearance that occurs in the ring-groove. With cast-iron pistons and rings the initial fit can be established fairly well and, under operating conditions, the expansion of the ring is practically the same as that of the groove; consequently, no additional clearance occurs until the ring or groove becomes warm. It is not possible to determine when increased clearance will occur, as this depends on the conditions under which the engine is operating. With a three-ring piston the actual clearance will vary from the top down in about the following relation: upper, 0.010 in.; second, 0.006 in.; and third, 0.003 in.

The extent of side-clearance unquestionably is the factor that is responsible for any difference between cast-iron and aluminum pistons. Regardless of the accuracy of the fit of the ring in the groove, the difference in expansion between the ring-groove and the ring on an aluminum piston produces initial clearance in excess of that required to effect the proper seal and increase in the rate of wear between the ring and groove because of the initial reciprocation. The use of oil-return holes in the ring-groove at the back of the ring is very beneficial when the ring has side-clearance so that the oil has access to the space back of the ring.

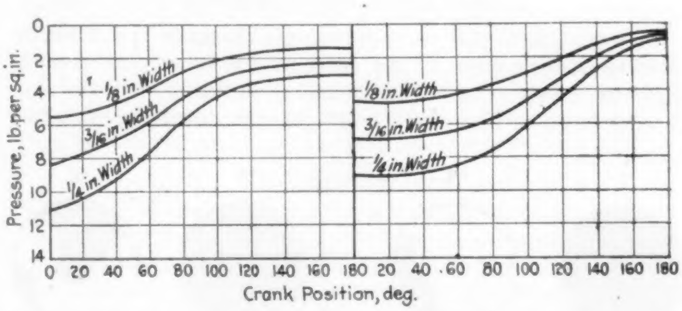


Fig. 8—Diagram Showing the Unit Pressure on the Ring-Groove Produced by Friction and Inertia Forces of Rings of Different Widths at 1000 R.P.M.

The two halves of Fig. 10 indicate the benefits of these oil-return holes. In the upper view the engine is operated at 500 r.p.m. and at 1000 r.p.m. in the lower, the equipment being the same. It is obvious that, with holes so located, particularly when used in a ring-groove above the piston-pin, the ring is useless for retaining the compression. Cylinders Nos. 1 and 2 each have one ring with 0.005-in. side-clearance, and oil-return holes in the ring-groove. Cylinder No. 3 has rings $\frac{1}{8}$ in. wide and no relief-holes. The duration of the test was 3 min. for the upper view and 1 min. for the lower. Fig. 10 should be compared with Fig. 9.

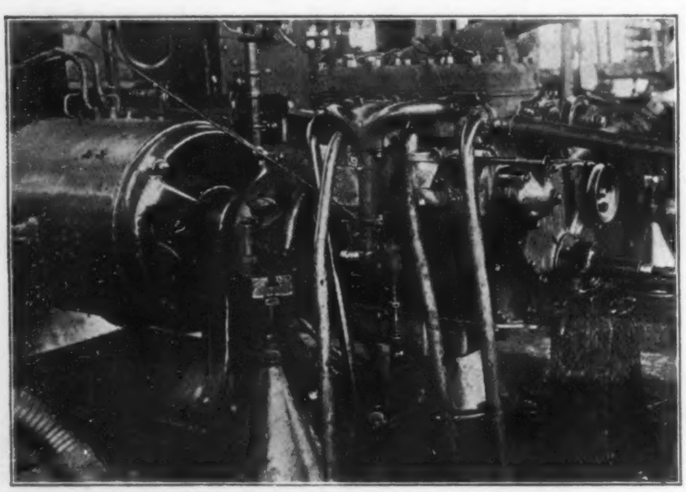


Fig. 7—Part of the Equipment for Measuring the Quantity of Oil Pumped Through the Exhaust

Thin Rings

The influence of ring thickness on side-clearance has received considerable attention, particularly for use with

aluminum pistons. Reduction in ring thickness unquestionably reduces the relative clearance. The weight of the ring itself is reduced, consequently decreasing the inertia effects of the ring. The relative forces with rings of different thicknesses are represented in Fig. 8. Fundamentally, however, they cannot solve the difficulty. A multiplicity of thin rings in a narrow groove would provide a good solution. They must be made of steel rather than cast-iron, and they must be hard so that they will not lap the bore and cause excessive wear. An effective unit-pressure of the ring against the wall could be well maintained and, likewise, the inertia effects of the portions of the ring would be dampened out by the oil-film between the several rings. Circumferential sealing also would be improved. Piston No. 3 in Fig. 10 is equipped with four 1/8-in. rings above the piston-pin without any relief-holes, from which it would seem that no benefits are obtained.

Combining all the essential requirements of a ring, it seems that a ring of two-piece construction capable of sidewise expansion is desirable. It is important, however, that sufficient wall pressure be provided; otherwise the benefits of the side-seal are lost with the inability of the ring to displace the oil from the cylinder wall properly.

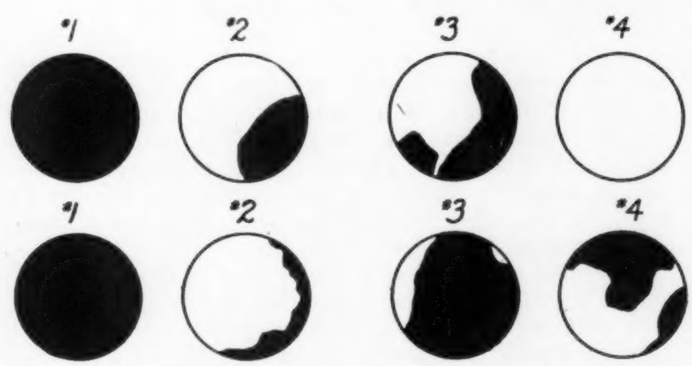


FIG. 9—TEST RESULTS OBTAINED (ABOVE) AT 500 R.P.M. AND (BELOW) AT 1000 R.P.M.

Cylinders Nos. 1 and 2 Had One Ring with a Side Clearance of 0.005 In., While the Other Two Had Standard Three-Ring Equipment. The Lower Speed Cylinder No. 1 Filled in 1 Min. and in 0.3 Min. at the Higher Speed. These Diagrams Should Be Compared with Those in Fig. 5

The clearance of the piston in the cylinder unquestionably has some effect on the functioning of the ring; however, this is not because an increased clearance between the piston and cylinder permits a larger quantity of oil to accumulate. From a fundamental standpoint the oil-film is not likely to be any thicker with a loose piston than a tight one. The oil pressure built up ahead of the ring, however, is increased with the closer fit because the oil-film is confined more closely.

Piston fit in the cylinder has an appreciable effect on the wear of the rings; on both the outside, which contacts with the cylinder wall, and the sides contacting with the ring-groove. It is evident that the piston will rock in the cylinder to the extent of the clearance, and that it tends to form a convex surface on the outside of the ring, completely destroying its oil-scraping efficiency. The continual sliding back-and-forth of the ring upon the seat wears the ring-groove very rapidly, and it is believed it is more responsible for wear than the actual pressure exerted by the ring on the groove because of its reciprocation.

On the basis that the ordinary ring is likely to carry up as much oil as it will scrape down, the use of the fourth ring below the piston-pin is of little advantage, except that it maintains its fit and condition very much better than the others and therefore has merit. Fundamentally, the use of a fourth ring placed at the lower end of the piston increases the amount of oil to be handled. It follows that the scraper-ring itself travels over a filmed surface equivalent to the stroke of the piston, while the upper rings also travel over a surface of similar length, the accumulated travel being greater and the oil consequently being relayed from one ring to the other. It is interesting to observe that, without any ring on the piston, absolutely no oil is passed up, the piston sliding over the lubricating film without disturbing it.

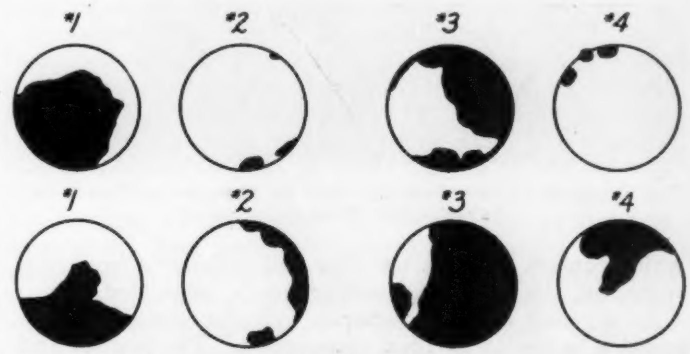


FIG. 10—HOW THE PRESENCE OF OIL-RETURN HOLES AFFECTS THE FORMATION OF OIL DEPOSITS

Cylinders Nos. 1 and 2 Were Equipped with One Ring Having a Side Clearance of 0.005 In. and Had Oil-Return Holes in the Ring-Groove, While Cylinder No. 3 Was Fitted with Rings 1/8 In. Wide and No Relief Holes. The Upper Drawing Shows the Results Obtained at 500 R.P.M. and the Lower One Those Obtained at 1000 R.P.M. These Results Should Be Compared with Those of Fig. 9

RING AND CYLINDER CONTACT EFFECTS

Plain rings change their position circumferentially in the cylinder and, if the cylinders are out-of-round, which is usual, due to unequal expansion and the like, considerable oil will be passed by the rings. It was evident in all the tests that have been conducted that the location of the ring circumferentially is very important and, in duplicating results, it is necessary to pin the rings so that they cannot rotate. To illustrate the extent to

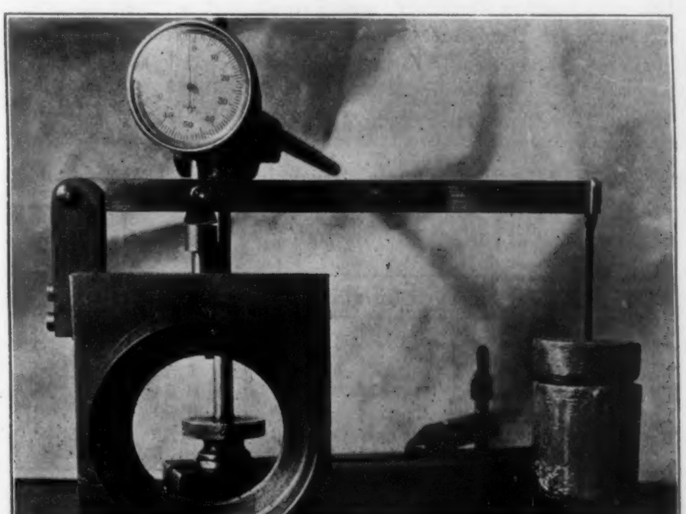


FIG. 11—APPARATUS USED FOR THE DETERMINATION OF PISTON-RING PRESSURE

With Multiple-Piece Rings, the Pressures Were Taken with the Ring Installed in the Groove

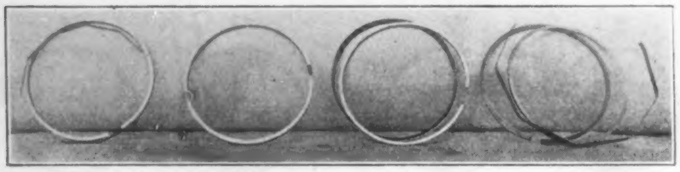


FIG. 12—SOME OF THE VARIOUS TYPES OF RING WITH WHICH THE EXPERIMENTS WERE CONDUCTED

which contact between the ring and cylinder affects consumption, the following comparison is submitted.

As a result of using separate cylinder sleeves, it was possible to obtain an ideal comparison. The pistons and rings remained undisturbed and new cylinder sleeves were used, the engine having seen considerable service. With old rings and cylinders, the oil consumption was 400 miles per gal. and with the same rings and pistons, but with new cylinders, the oil consumption was 150 miles per gal.

Summarizing the fundamental factors of the piston ring, we conclude that

(1) Drain-holes are absolutely essential
(2) An angular-faced piston-ring is beneficial
(3) The proper mechanical fit between the ring and the groove is essential
(4) Oil relief-holes in the rear groove are beneficial when side-clearance occurs
(5) The slot in the rear is unimportant
(6) The number of rings is unimportant

Fig. 11 shows the apparatus used for the determination of piston-ring pressure. With multiple-piece rings, the pressures were taken while the ring was installed in the groove.

Fig. 12 shows some of the various types of ring with which the experiments were conducted, and Fig. 13 illustrates several types of piston that were installed during the tests. Pistons having one ring-groove only were employed in obtaining the data on ring characteristics.

Table 3 gives piston-ring pressures taken in the groove and taken free. Table 4 shows the relative unit wall-pressure with change in contact area and face width.

Passing from a consideration of the mechanical aspects of rings and pistons in the controlling of oil passing to the combustion-chamber, mention should perhaps be made of some of the factors affecting carbonization and the

TABLE 3—PISTON RING PRESSURES

Ring Diameter, Outside, 3½ in.; Ring Width, 3/16 in.

Kind of Ring	Weight in Pounds, at Deflection of 0.001 in.	0.002 in.	0.003 in.
Weight Taken on Ring Not Installed in Groove			
Concentric, Hammered	4.96	6.10	6.650
Two-Piece, Outer	1.28	1.52	1.850
Two-Piece, Inner	.....	0.64	0.775
Three-Piece, Spring Type, Outer	0.00	0.00	0.000
Three-Piece, Spring Type, Inner	0.00	0.52	0.640
Weight Taken with Rings Installed in Groove			
Hammered	5.20	6.40	7.000
Two-Piece	3.00	3.65	4.400
Three-Piece, Spring Type	6.40	10.40	13.200
Hammered, Spring Type	15.90	24.52	33.000

TABLE 4—RELATIVE WALL PRESSURE OF RING OF 3½-IN. OUTSIDE DIAMETER

Outside Diameter of Ring, 3½ in.			Deflection	0.001 in.	0.001 in.
Ring Width, In.	Normal Ring Tension, Lb.	Developed Contact Area, Sq. In.	Wall Pressure, Lb. per Sq. In. Normal	Tight	Loose
3/16	4.5	0.665	6.75	8.25	5.25
1/8	4.5	0.404	11.20	13.60	8.65
3/32	4.5	0.341	13.20	16.20	10.20

fouling of spark-plugs after the oil has reached the combustion-chamber. First, of course, is the nature of the lubricant. It usually is considered that the heavier oil produces more carbon deposit, although this is not always true. The extent of the free carbon in the lubricant has little effect upon combustion-chamber deposits. In this respect the benefits of proper heat conductivity on the piston-head, to prevent excessive carbon deposit on the inside, should be observed. The location of the spark-plug is of great importance, as in many cases, of course, frequent fouling of the spark-plugs has been a result of improper location, the least evidence of over-oiling immediately affecting its operation.

A typical condition of cylinder-head carbon-deposit is shown in Fig. 14. The maximum deposits are on the side of the cylinder upon which most of the oil is thrown from the crank, the spark-plugs being sufficiently removed from this zone to prevent their becoming carbonized.

CONTROL OF THE OIL SUPPLY

Control of the oil supply, particularly that from the bearings to the cylinders, unquestionably has more influence on the oil consumption than the mechanical condition of the pistons and the rings. Stated briefly, it is very evident that, under the best conditions, the rings and the pistons can only exercise a very definite control over the amount pumped to the combustion-chamber.

In the discussion of the mechanical aspect of the pistons and the rings, it is concluded that there are definite fundamental conditions that must exist to control the oil

FIG. 13—SEVERAL TYPES OF PISTON INSTALLED DURING THE TESTS
Pistons Having One Ring-Groove Only Were Employed in Obtaining the Data on Ring Characteristics

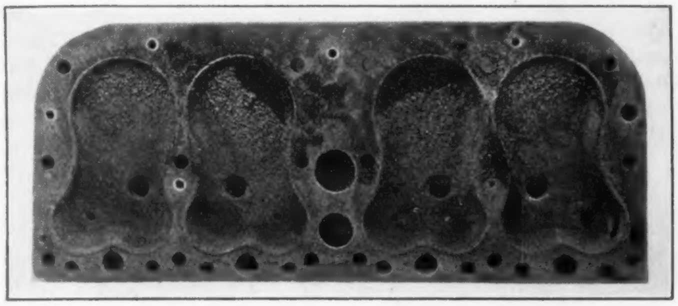


Fig. 14—A Typical Cylinder-Head Showing the Carbon Deposit in a Tractor Engine

passing to the combustion-chamber to the best extent, and the problem of the piston-rings and the pistons is to maintain these fundamental conditions. This same argument holds for the control from the source of supply. It is probable that, with the very close fits of bearings, the extent of oil discharge to the cylinders will be of such magnitude that the rings can control it properly. As wear of the bearings occurs, the discharge is increased greatly and this is true particularly where an oil-regulating mechanism is provided that will maintain the oil pressure even though the bearings become loose.

It will be further evident from the analysis yet to be made that, in most instances, the effect of higher speed is to increase the quantitative discharge from the bearings and it will be shown that, while ordinarily higher speed will increase the oil consumption considerably, it is believed that this is not due to any change in functioning of the piston-rings in keeping the oil from passing into the combustion-chamber. In other words, the rings will be almost equally efficient within the ranges of piston speed under average conditions, if they are fitted properly.

Trucks and tractor engines that operate a large percentage of the time at governed speeds are affected greatly in their oil consumption by the bearing condition. It is also true of passenger-car engines that a car will show a reasonable economy when operated at ordinary driving speeds and, while this economy may be more or less superficial as a result of the dilution that takes place, the same car running at higher speeds, where dilution does not become a factor, will consume unreasonable quantities.

The consideration of the oil discharge from the crankcase is, naturally, confined largely to pressure-feed systems. With splash feed for the connecting-rods, the amount thrown onto the cylinder wall depends, of course, on the dip. It is not controlled as easily except through the use of baffle-plate. In most instances pressure-feed crankshafts are provided with an oil-hole in the crankpin radially outward, in which position the extent of the oil discharge is affected greatly by bearing clearances.

Let us refer for a moment to two diagrams in Fig. 15 which indicate the pressure distribution on the crankpin bearing throughout the cycle at a speed of 500 r.p.m. It is evident that, throughout the greater portion of the cycle, the pressure on the pin is radially outward in the region of the inside of the crankpin, thus locating the clearance in the bearing on the outside. As it approaches the center on the compression stroke, and likewise on the expansion stroke, the pressure is reversed.

Fig. 16 gives the pressure distribution at 1500 r.p.m. under full load. The increase in the magnitude of the inertia and centrifugal forces, as the speed increases, is such that the resultant pressure on the pin is radially outward except for an angular travel of 100 deg. during the expansion stroke. It is evident from these figures that, at all speeds and particularly as the speed increases, the pressure on the pin is concentrated chiefly on the inside, thus locating all of the bearing clearance on the outside of the pin. When running under no load, this condition is emphasized. It is clear that, with the oil-hole placed on the outside of the crankpin, the clearance of the bearing will be located in the vicinity of the hole most of the time, thus permitting a free opening through which the oil can be discharged. The oil discharged from the crankpin has a direction tangentially forward from the point of discharge; so, it will be evident when approaching the upper center that the oil discharged from the crankpin is directly in line with the cylinder. Bearing wear and increased clearance will therefore have a marked effect on the quantity of oil thrown from the bearing and, from the figures that follow, it points very conclusively to the influence bearing clearance has on oil-pumping and oil consumption.

Fig. 17 is a reproduction of similar pressure-diagrams

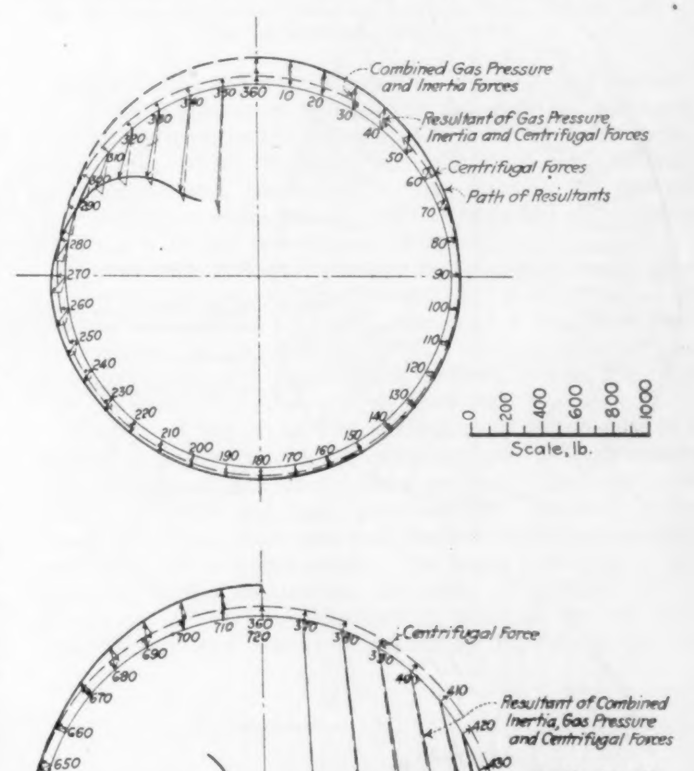


Fig. 15—Crankpin Bearing Pressures at a Speed of 500 R.P.M. The Upper Diagram Is for the Intake and Compression Strokes; the Lower for the Expansion and Exhaust Strokes

TABLE 5—OIL CONSUMPTION

Car No.	Type of Rings	Oil-Hole Location	Total Miles Run	Speed, m.p.h.	Oil Consumption, Miles per Gal.: Total	Distributing Hole: Outside	Distributing Hole: Inside
Series No. 1							
Experimental	Standard, Old	0	50	40	76.00	......	......
Experimental	New	0	50	40	143.00	......	......
Experimental	New, 1/8 In.	0	50	40	64.00	......	......
Experimental	Special	0	50	40	221.34	......	......
Series No. 2							
G	Standard, Old	0	50	40	139.00	......	......
G, Trial No. 1	Special	0	50	40	300.00	......	......
G, Trial No. 2	Special	0	50	40	240.00	......	......
G	Standard, Old	1	50	40	755.00	......	......
Series No. 3[e]							
58,301	....	..	50	40	......	156	382
O. B.	....	..	50	40	......	151	388
60,887	....	..	50	40	......	156	692
65,629	....	..	50	40	......	294	512

[e] In this series only the changes in the oil distribution were considered, losses from other sources being neglected.

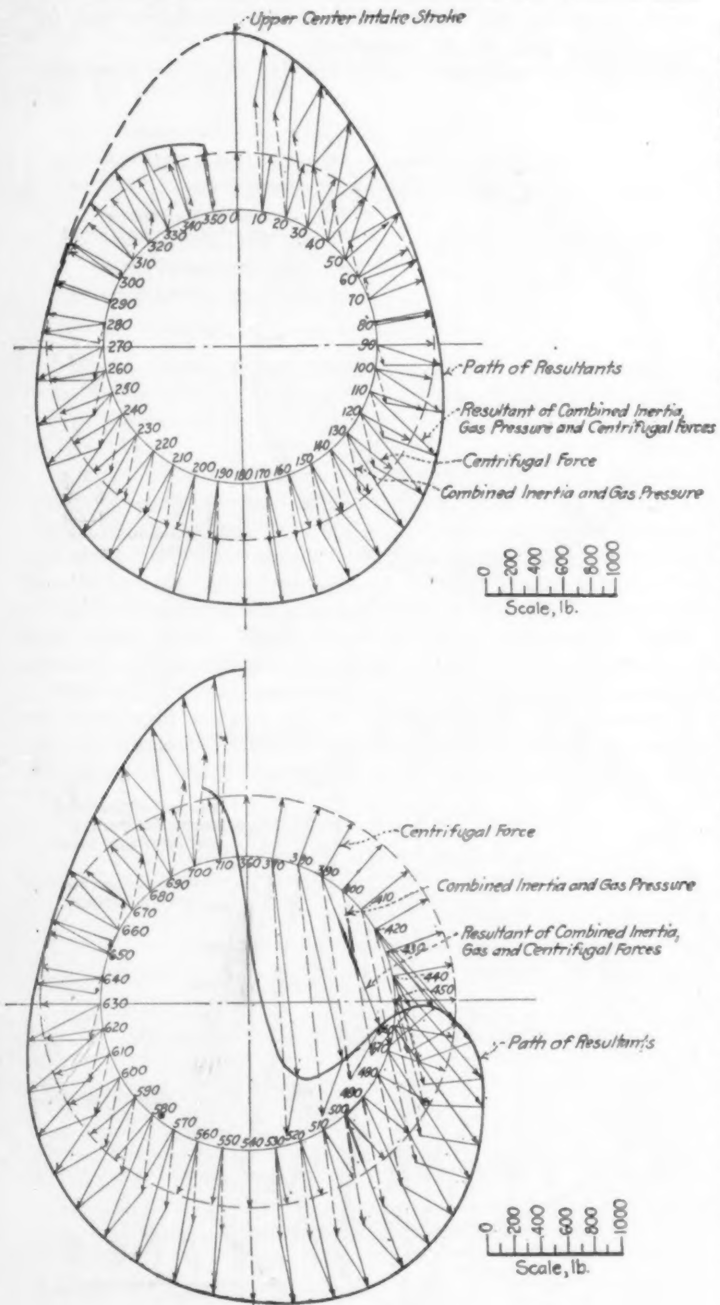


FIG. 16—CRANKPIN BEARING PRESSURES AT A SPEED OF 1500 R.P.M. The Upper Diagram Is for the Intake and Compression Strokes; the Lower for the Expansion and Exhaust Strokes

of eight-cylinder V-type engines, for which the same arguments hold true. To control the extent of oil discharged to the crankpin, it is clear that the oil-hole should be located to maintain a continuous seal regardless of the bearing clearance and the speed; therefore, it should be in the zone of continued pressure, preferably radially inward on the inside of the crankpin.

To appreciate the influence of oil-hole location on the extent of oil-pumping and oil consumption, compare cylinders Nos. 3 and 4 in the preceding figures, that have standard ring equipment. In all the preceding tests the bearing clearance was increased purposely to approximate service conditions. The comparative quantity of oil as measured from the exhaust discharge or as clearly pictured in the preceding diagrams between cylinders Nos. 2 and 4 and Nos. 1 and 3 is very marked. Fig. 9 shows this condition particularly well. With the same piston-ring equipment in cylinders Nos. 1 and 2, cylinder No. 1 is considerably more affected by a change in the rings. The influence of speed can be observed also. Piston No. 1 becomes noticeably worse, while piston No. 2 is not affected appreciably.

The tests reported in Table 5 are submitted in further support of the controlling influence of crankpin discharge, and the effects of special piston-ring equipment can be observed also. In series No. 3 the oil consumption obtained at 40 m.p.h. with standard ring equipment and the outside oil-hole on the crankpin is given, and also the oil consumption in miles per gallon obtained with the oil-hole on the inside, all other conditions remaining the same. Series No. 3 should be compared with Series No. 1, which shows differences in ring equipment. The use of special ring equipment effected an improvement in some cases but, upon succeeding tests, the oil consumption would increase gradually.

It is interesting to note that increasing the load has the effect of establishing the bearing clearance in the vicinity of the oil-hole when located on the inside through a part of the expansion stroke and, consequently, it exercises a very favorable control on the amount of oil supplied in proportion to the load. From the standpoint of effective lubrication of the pin some arguments can be advanced but, from a consideration of Fig. 18, it would appear that better oil-film distribution is obtained with the inside location of the oil-hole, the influence of centrifugal force tending to distribute the film around the bearing, while with the outside oil-hole the oil is immediately thrown from the bearing. Actual experience has proved that bearing failures at high speed are less pronounced with the inner location of the hole, and that dirt will not score the bearing so readily. Efforts to time

the oil supply to the crankshaft through the main bearings properly, so that the discharge from the pin would not occur when the shaft was passing the cylinder, proved unsuccessful, because the quantity of oil in the channels of the shaft disturbed the control. The effect of main-bearing discharge should be considered also. It

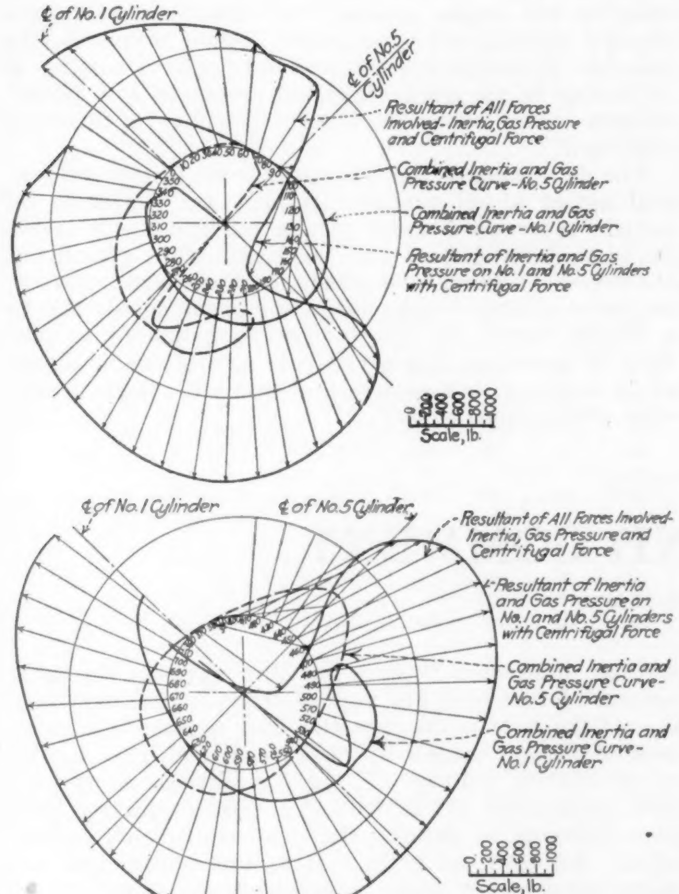


FIG. 17—CRANKPIN BEARING PRESSURES IN AN EIGHT-CYLINDER 90-DEG. ENGINE AT 2000 R.P.M.
The Upper Diagram Is for the Intake and Compression Strokes; the Lower for the Expansion and Exhaust Strokes

is of advantage to provide a thrower flange adjacent to the rear bearings so that the oil cannot reach the crankpin hole and, consequently, be discharged directly into the cylinder.

An interesting test was conducted to determine the relation between pressure feed and splash feed. An engine was converted to use a splash-feed system, retaining the same piston equipment. It was observed that almost any result could be obtained with the splash feed and, if adequate lubrication were provided for the pin, an excessive quantity was thrown into the cylinders unless a baffle-plate was used. The troughs into which the rods dipped were made so that they could be raised and lowered as desired.

Influence of Oil Viscosity and Effects of Dilution

A reduction in oil viscosity as a result of an increase in oil temperature, dilution or the use of light oils, will increase proportionately the oil consumption by reason of the additional discharge from the crankpin under the same pressure, unless the oil-hole is located in the zone of pressure so that it exercises a definite control. With an engine so equipped, practically no difference is ob-

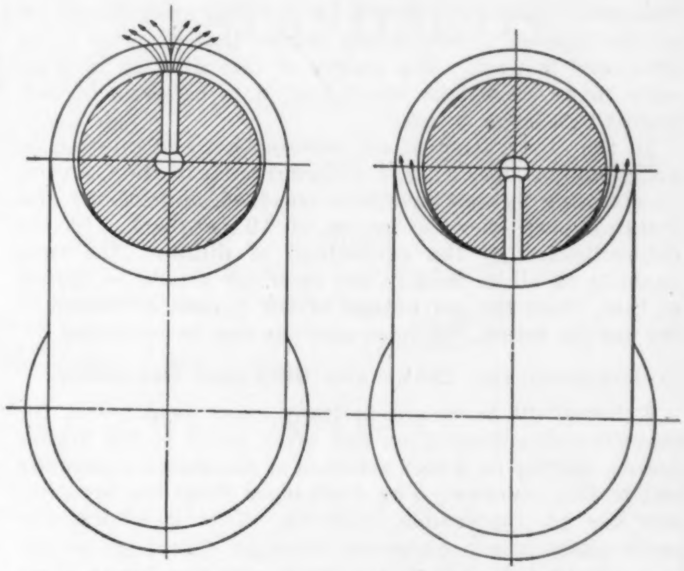


FIG. 18—OIL DISCHARGE FROM A CRANKPIN WITH INSIDE AND OUTSIDE OILHOLES RESPECTIVELY

tained with the different viscosities produced by the control of the temperature when using the same oil, although a slight addition in the consumption is shown with light-bodied oils, which indicates the effect of cylinder-wall temperature and, consequently, the viscosity of the oil on the cylinder walls and the difference in the piston-ring efficiency with the lowered viscosity.

Fig. 19 indicates the relative oil consumption in proportion to viscosities with different oils, *A* being an oil of 500-sec. viscosity at 104 deg. fahr. and *B* an oil of 650-sec. viscosity at 104 deg. fahr. The curve *C* is a portion of a curve prepared by C. W. Stratford, giving the relation between viscosity and oil consumption.

Dilution of the oil by the admixture of liquid fuel is a problem associated with oil consumption. As previously stated, dilution may occur under ordinary operating conditions to the extent that, quantitatively, the fluid in the crankcase is not decreased and the operator assumes that very little oil is being used. The rapid lowering of the viscosity is the dominating influence of dilution which, of course, affects the lubricating value of the oil very considerably and makes it necessary to change the oil

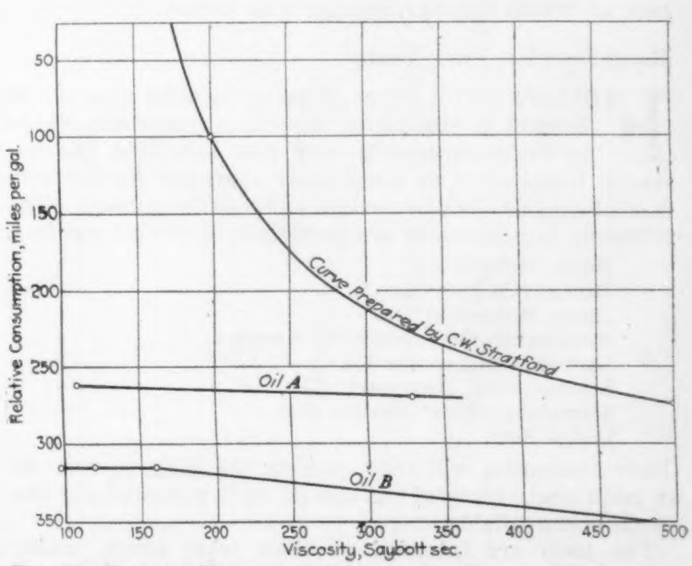


FIG. 19—DIAGRAM SHOWING THE OIL CONSUMPTION WITH RELATION TO THE VISCOSITY AND THE CHARACTER OF THE OIL

frequently to obtain efficient lubrication, unless some one of the types of crankcase-oil refiner that are now being developed is used. The extent of this dilution depends upon many conditions which can, it is believed, be controlled to a large extent.

In the observation of oil consumption and dilution on many engines operating in different parts of the Country, no alarming figures have been obtained, and in very few instances are they in excess of 10 per cent. In the determination of the percentage of dilution, the total quantity of oil or fluid in the reservoir should be known so that, from the percentage of the diluent contained in the sample tested, the total quantity can be computed.

External Oil Leaks and Breather Discharge

External oil leaks are in many cases responsible for excessive oil consumption and often occur at the higher speeds, leaving no direct evidence of the quantity actually lost in this manner. Fog discharged from the breather also has an appreciable influence. This is emphasized particularly where excessive crankpin discharge occurs and also in splash-feed-lubrication engines where light oils are used. Under the latter condition the oil becomes fogged quickly and, if accompanied by piston blow-by, oil vapor will be emitted from the breather.

Controlling the oil pressure in proportion to the throttle opening to effect proportionate lubrication and obtain satisfactory oil economy does not seem to exercise a definite control unless proper consideration is given also to the location of the oil-hole in the crankpin. It is clear that, at the higher speeds, with the increase in the throttle opening and consequently higher pressures, the crankpin discharge will be increased proportionately. It is feasible to use much higher oil pressures with proper oil-hole control which in itself constitutes a favorable argument.

The data presented in this paper cover briefly the general aspect of oil consumption and are supported by actual observations under service conditions. It is believed that the pistons and the rings, when embodying the necessary fundamentals will, in themselves, exercise control over the oil-pumping at the lower speeds and to a definite extent. To insure oil economy under all conditions of operation, it is essential to control the oil supply at its source with a construction that will maintain constant effectiveness.

CHRONICLE AND COMMENT

(Concluded from page 444)

In a project and progress chart of highway investigations prepared recently by Doctor Hatt, the four principal classifications are economics of grades, tractive resistance, capital and operating vehicle costs, and construction and maintenance highway costs. Various items of work subsidiary to these classifications are now under way at Iowa State College, University of Michigan, Yale University and Massachusetts Institute of Technology. The matters in connection with which the members of the Society in general are best qualified to serve relate to the establishing of basic items of cost for motor vehicles, including the determination of capital and operating costs for motor trucks, passenger cars and motorbuses. It is expected that there will be general discussion of these subjects at the meeting of the Society to be held at White Sulphur Springs this month.

Road-Service Fuel Tests

FOLLOWING a series of tests, recently run by the Bureau of Standards, showing a consistent change in fuel-consumption with fuel volatility, the Research Department of the Society arranged for the conduct of similar tests by several automobile builders. The following organizations are participating in this work:

- Buick Motor Co.
- Packard Motor Car Co.
- Hupp Motor Car Co.
- Studebaker Corporation of America
- Chevrolet Motor Car Co.
- International Harvester Co.
- Stromberg Motor Devices Co.
- Dodge Bros.

Other companies will start making the tests as soon as an additional supply of the special fuels prepared for the purpose is available.

The tests are intended to show what effect, under normal everyday running conditions with average drivers, distinct differences in fuel volatility have on the total amount of fuel used per mile of travel. The main feature of the tests is the operation of a number of cars of each of several models by their regular drivers in the course of their ordinary driving, but each car being supplied successively for periods of 1 week's duration with fuels differing by definite steps in volatility characteristics. An essential point is that the drivers are not informed as to the grade of fuel they are using at any time. The tests will not interfere in the slightest degree with the regular service of any of the cars further than to permit the filling and draining of the fuel-tanks during the test for checking the consumption.

Copies of all the records are to be forwarded to the Society's Research Department for compilation and comparison. The final results of the tests are to be used to estimate the effect of fuel changes on the total fuel-consumption of the Country.

A joint test program, including elaborate road tests of vehicles and far-reaching fuel investigations, is to be conducted by the Bureau of Standards, the Bureau of Mines, and other government agencies, with the support and cooperation of the National Automobile Chamber of Commerce and the Society of Automotive Engineers, representing the automotive industries, and the American Petroleum Institute, representing the petroleum industry. The tests are under the general supervision of the Research Department of the Society of Automotive Engineers and the Research Division of the American Petroleum Institute.

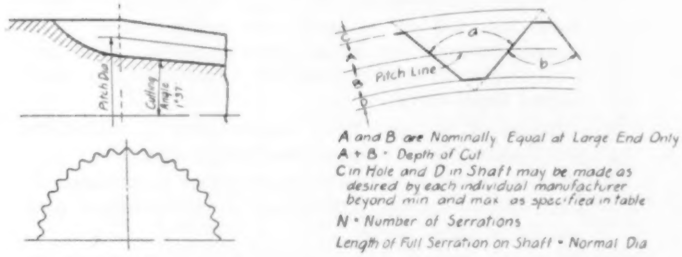
When the tests are completed, and the information derived has been compiled, it will be possible to state more exactly what grade of gasoline should be distilled and sold to the public to conserve the petroleum resources of the Country, and at the same time provide the public with a grade of gasoline that will give good mileage at a reasonable price.

REPORTS OF DIVISIONS TO STANDARDS COMMITTEE

(*Concluded from page 488*)

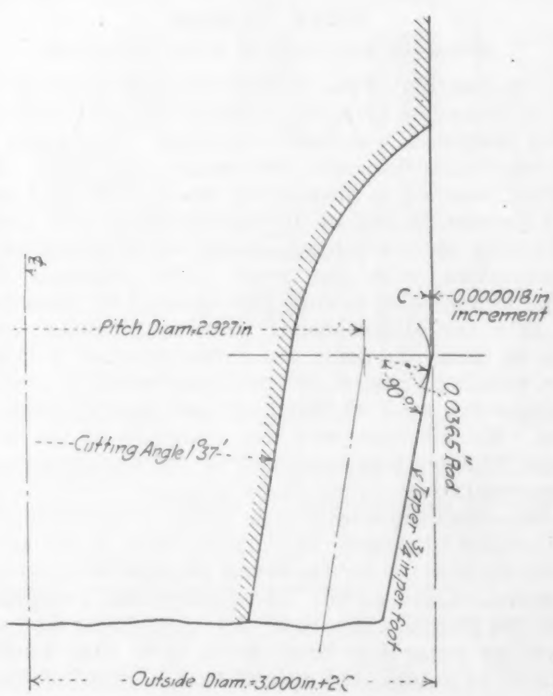
Nominal Diam.	Pitch Diam.		N	a, deg.	b, deg.	HOLE			SHAFT		
						Large Diam. Min.	Small Diam.		Outside Diam.		Inside Diam. Max.
	Max.	Min.					Max.	Min.	Max.	Min.	
½	0.485	0.483	36	90	80	0.500	0.469	0.468	0.499	0.498	0.467
⅝	0.605	0.603	36	90	80	0.625	0.584	0.583	0.624	0.623	0.582
¾	0.733	0.731	36	90	80	0.750	0.716	0.714	0.749	0.747	0.713
⅞	0.855	0.853	36	90	80	0.875	0.835	0.833	0.874	0.872	0.832
1	0.977	0.975	36	90	80	1.000	0.954	0.952	0.999	0.997	0.951
1⅛	1.098	1.096	36	90	80	1.125	1.071	1.069	1.124	1.122	1.068
1¼	1.220	1.218	36	90	80	1.250	1.190	1.188	1.249	1.247	1.187
1⅜	1.342	1.340	36	90	80	1.375	1.309	1.307	1.374	1.372	1.306
1½	1.464	1.462	36	90	80	1.500	1.428	1.426	1.499	1.497	1.425
1¾	1.708	1.706	36	90	80	1.750	1.666	1.664	1.749	1.747	1.663
2	1.952	1.949	48	90	82½	2.000	1.904	1.902	1.999	1.997	1.901
2¼	2.196	2.193	48	90	82½	2.250	2.142	2.140	2.249	2.247	2.139
2½	2.440	2.437	48	90	82½	2.500	2.380	2.378	2.499	2.497	2.377
2¾	2.684	2.681	48	90	82½	2.750	2.618	2.616	2.749	2.747	2.615
3	2.928	2.925	48	90	82½	3.000	2.856	2.854	2.999	2.997	2.853

All dimensions in inches and apply to large end of taper only.
Taper ¾ in. per ft. on outside diameter.
Cutting Angle 1 deg. 37 min.



of gasoline-tank filler-pipe openings on automobiles and motor-trucks. This matter was referred to the Parts and Fittings Division for consideration in connection with the present S.A.E. Recommended Practice for Tank and Radiator Caps, page 650 of the S.A.E. HANDBOOK. It developed that the filler-pipe openings are in most cases 2 in. or larger in diameter and that a 2-in. minimum opening is acceptable. Therefore

The Parts and Fittings Division recommends that the present S.A.E. Recommended Practice for Tank and Radiator Caps, page C58 of the S.A.E. HANDBOOK, be extended by adding the note, "It is recommended that on passenger-cars, motor-trucks and tractors gasoline-tank filler-pipes have a minimum clear opening of 2 in. in diameter"



Lock-Washers

(Proposed Extension of S.A.E. Standard)

At the meeting of the Parts and Fittings Division on May 9 attention of the members was called to the trouble that had been experienced due to the lack of temper or to the brittleness of the lock-washers. The members of the Division felt that the temper and toughness tests which were included in the S.A.E. Standard previous to the adoption of the last revision, should be included in the present standard to provide an adequate check on the quality of lock-washers in guarding against the troubles that have been experienced. Therefore

The Parts and Fittings Division recommends that the present S.A.E. Standard, page C5 of the S.A.E. HANDBOOK, be extended to include the accompanying temper and toughness tests.

Temper Test. — After compressing the lock-washer to flat, the reaction shall be sufficient to indicate necessary spring power, and on a subsequent compression to flat the lock-washer shall manifest no appreciable loss in reaction

Toughness Test.—Forty-five per cent of the lock-washer, including one end, shall be secured firmly in a vise, and 45 per cent, including the other end, shall be secured firmly between parallel jaws of a wrench. Movement of wrench at right angles to the helical curve shall twist the lock-washer through 45 deg. without it showing signs of fracturing, and shall twist the lock-washer entirely apart within 135 deg.

SCREW THREADS DIVISION REPORT

Division Personnel

E. H. Ehrman, *Chairman*	Chicago Screw Co.
O. B. Zimmerman, *Vice-Chairman*	International Harvester Co.
Earle Buckingham	Pratt & Whitney Co.
E. Burdsall	Russell, Burdsall & Ward Bolt & Nut Co.
Luther Burlingame	Brown & Sharpe Co.
G. S. Case	Lamson & Sessions Co.
W. R. Mitchell	National Acme Mfg. Co.
Alex Taub	General Motors Corporation

Screw Threads

(Proposed Extension of S.A.E. Standard)

At the meeting of the Miscellaneous Division in Cleveland in December 1916 the proposal to standardize screw-thread tolerances was first considered. Not much progress was made, however, for some little time. E. H. Ehrman, who was a member of the Division and also of the American Society of Mechanical Engineers Committee working on this subject, conducted a number of tests in connection with the work. The National Screw Thread Commission having been created by Congress in July 1918 the Miscellaneous Division concluded that it would be better to defer the formulation of a Division report pending a report by the Commission, in order not to parallel the work of that body and possibly cause confusion. Mr. Ehrman, who was chairman of the Miscellaenous Division, was appointed by the Society as one of its representatives on the Commission.

In May 1919 the Commission issued a tentative report that included in general the classification of fits and the detailed dimensions for the screw threads included under the several classes of fit. The Society was requested to review the progress report of the Commission and make suggestions regarding those parts of it that would be applicable to automotive apparatus. A meeting held in New York City of Sept. 8, 1919, was attended by representatives of screw-thread parts producers and users, the American Society of Mechanical Engineers, the Bureau of Standards and members of the National Screw Thread Commission. General discussion of the Commission's progress report indicated the desirability of further investigating the application of its provisions in the automotive industries, and a special Society Committee was appointed for this purpose, consisting of

Paul W. Abbot, *Chairman*	Lincoln Motor Co.
W. K. Jamison	Domestic Engineering Co.
B. P. Smith	Packard Motor Car Co.
Lyle K. Snell	Willys-Overland Co.
Alex Taub	General Motors Corporation

This Committee wrote more than 200 representative companies engaged in automotive work for data as to their current screw-thread practice. The Society was assisted in securing returns to the questionnaire by the National Automobile Chamber of Commerce, the Motor & Accessory Manufacturers Association and the National Gas Engine Association.

The special Committee issued its report to the Society on Jan. 3, 1920, and it was transmitted to and duly considered by the National Screw Thread Commission in preparing its progress report of June 1920.

The Miscellaneous Division of the Standards Committee had been kept informed by its chairman and the Society of the progress that had been made in this work. Meantime a Sectional Committee had been formed, which was sponsored by the Society and the American Society of Mechanical Engineers for further study of the National Screw Thread Commission's progress report toward its adoption by the mechanical industries in general. The Division felt that the Society should not adopt specific data in connection with screw-thread fits and dimensions until the work in progress by the Sectional Committee indicated approval of the National Screw Thread Commission's report. The Sectional Committee referred the report of the Commission to a Working Committee of the Sectional Committee, which reported in February 1922, having made the following selections from and changes in the Commission's report:

(1) Certain combinations of thread series and fit have been selected as suitable for general use. The coarse-thread close fit has been omitted. It is recommended that the loose-fit tolerances be published separately

(2) Classes of fit have been renumbered to eliminate the Subdivision of the Commission's Class II

(3) The material relating to gaging has been omitted, with the recommendation that it be developed and published separately

(4) The pipe-thread standard has been omitted, since it has already been approved as an American standard and published by the A.E.S.C.

(5) Fire hose and regular hose standards have been omitted. The publication of these as a separate pamphlet is recommended

(6) The Fine-Thread Series (S.A.E.) has not been carried beyond 1½ in. 12 pitch

(7) The relation between lead or form errors and pitch-diameter tolerances has been treated in detail

(8) Extreme or drawing tolerances only are given

(9) The material has been generally condensed and somewhat rearranged

In January 1920 the Screw Threads Division of the Standards Committee was organized to consider all

screw-thread standardization matters for the Society, and took over the former work of the Miscellaneous Division. No separate action was taken on the work before the National Screw Thread Commission by the Screw Threads Division toward establishing an S.A.E. Standard for screw-thread dimensions, pending the conclusion of the National Screw Thread Commission and the Sectional Committee's work. With the completion of the report of the Working Committee of the Sectional Committee in February of this year, the Division felt that the Society could properly proceed to adopt the classification of fits and tables of dimensions for screw threads which are used by the automotive industries, and now recommends that the following definitions and tables, taken from the report of the Working Committee of the Sectional Committee, be adopted as S.A.E. Standard.

The Working Committee's report is still to be approved by the Society and the American Society of Mechanical Engineers as sponsors for the Sectional Committee, organized under the rules of the American Engineering Standards Committee, but the Division members feel that the following recommendations can now be adopted by the Society. Therefore

The Screw Threads Division recommends that the present S.A.E. Standard for Screw-Threads, page C1 of the S.A.E. HANDBOOK, be extended to include the accompanying Tables Nos. 1 to 7 inclusive, together with the definitions.

Class II B—Medium Fit. — This class of screw-threads shall be defined and specified as follows:

(1) Minimum Nut is Basic

(2) Maximum Screw is Basic

(3) Direction of Tolerance on Nut. The tolerance on the nut shall be plus

(4) Direction of Tolerance on Screw. The tolerance on the screw shall be minus

(5) Zero Allowance. The allowance between the pitch diameter of the maximum screw and the minimum nut shall be zero for all pitches and all diameters

(6) Tolerance Values. The tolerances for a screw or nut of a given pitch shall be as specified in table 3.

Class II A—Free Fit.—This class of screw-threads shall be defined and specified as follows:

(1) Minimum Nut is Basic

(2) Maximum Screw is Basic

(3) Direction of Tolerance on Nut. The tolerance on the nut shall be plus

(4) Direction of Tolerance on Screw. The tolerance on the screw shall be minus

(5) Zero Allowance. The allowance between the

TABLE 1—ALLOWANCES AND TOLERANCES FOR SCREWS AND NUTS, CLASS IIA, FREE FIT

Threads Per In.	Allowance, In.	Pitch-Diameter Tolerances, In.	Lead,[1] In.	Half Angle, min. sec.
80	0.0000	0.0017	0.0005	2—50
72	0.0000	0.0018	0.0005	2—41
64	0.0000	0.0019	0.0005	2—23
56	0.0000	0.0020	0.0006	2—11
48	0.0000	0.0022	0.0006	2—04
44	0.0000	0.0023	0.0007	2—03
40	0.0000	0.0024	0.0007	1—52
36	0.0000	0.0025	0.0007	1—49
32	0.0000	0.0027	0.0008	1—41
28	0.0000	0.0031	0.0009	1—45
24	0.0000	0.0033	0.0009	1—32
20	0.0000	0.0036	0.0010	1—24
18	0.0000	0.0041	0.0012	1—26
16	0.0000	0.0045	0.0013	1—24
14	0.0000	0.0049	0.0014	1—20
13	0.0000	0.0052	0.0015	1—19
12	0.0000	0.0056	00.016	1—18
11	0.0000	0.0059	0.0017	1—15
10	0.0000	0.0064	0.0018	1—14
9	0.0000	0.0070	0.0020	1—13
8	0.0000	0.0076	0.0022	1—11
7	0.0000	0.0085	0.0024	1—09
6	0.0000	0.0101	0.0029	1—10
5	0.0000	0.0116	0.0033	1—07
4½	0.0000	0.0127	0.0037	1—06
4	0.0000	0.0140	0.0040	1—05

[1] Variation in lead between any two threads not farther apart than the length of engagement.

The tolerances specified for the pitch diameters are cumulative and include all errors of lead and angle. The full tolerance on the pitch diameter is therefore not available unless the lead and the angle of the thread are perfect. The last two columns give as general information the error in lead, per length of thread engaged, and in angle respectively that can each be compensated for by half the tolerance on the pitch diameter. If the lead and the angle error both exist to the amount tabulated, the pitch diameter of a bolt, for example, must be reduced by the full tolerance or it will not enter a basic nut or gage. If no lead error existed on such a bolt, the angle error could be twice that given, and conversely; but these extreme conditions are not contemplated as being desirable.

TABLE 2—ALLOWANCES AND TOLERANCES FOR SCREWS AND NUTS CLASS IIB, MEDIUM FIT

Threads Per In.	Allowance, In.	Pitch-Diameter Tolerances, In.	Lead,[1] In.	Half Angle, min. sec.
80	0.0000	0.0013	0.0004	2—07
72	0.0000	0.0013	0.0004	1—54
64	0.0000	0.0014	0.0004	1—49
56	0.0000	0.0015	0.0004	1—41
48	0.0000	0.0016	0.0005	1—32
44	0.0000	0.0016	0.0005	1—24
40	0.0000	0.0017	0.0005	1—21
36	0.0000	0.0018	0.0005	1—17
32	0.0000	0.0019	0.0005	1—12
28	0.0000	0.0022	0.0006	1—13
24	0.0000	0.0024	0.0007	1—08
20	0.0000	0.0026	0.0008	1—02
18	0.0000	0.0030	0.0009	1—04
16	0.0000	0.0032	0.0009	1—01
14	0.0000	0.0036	0.0010	1—00
13	0.0000	0.0037	0.0011	0—57
12	0.0000	0.0040	0.0012	0—57
11	0.0000	0.0042	0.0012	0—55
10	0.0000	0.0045	0.0013	0—54
9	0.0000	0.0049	0.0014	0—52
8	0.0000	0.0054	0.0016	0—51
7	0.0000	0.0059	0.0017	0—49
6	0.0000	0.0071	0.0020	0—50
5	0.0000	0.0082	0.0024	0—48
4½	0.0000	0.0089	0.0026	0—47
4	0.0000	0.0097	0.0028	0—46

[1] Variation in lead between any two threads not farther apart than the length of engagement.

The tolerances specified for the pitch diameters are cumulative and include all errors of lead and angle. The full tolerance on the pitch diameter is therefore not available unless the lead and the angle of the thread are perfect. The last two columns give as general information the error in lead, per length of thread engaged, and in angle respectively that can each be compensated for by half the tolerance on the pitch diameter. If the lead and the angle error both exist to the amount tabulated, the pitch diameter of a bolt, for example, must be reduced by the full tolerance or it will not enter a basic nut or gage. If no lead error existed on such a bolt, the angle error could be twice that given, and conversely; but these extreme conditions are not contemplated as being desirable.

TABLE 3—CLASS IIA, FREE FIT, FOR SCREWS OF THE COARSE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Max.[1]	Tolerance	Min.	Max.[1]	Tolerance[2]	Min.	Max.[3]	Tolerance	Min.
1	64	0.0730	0.0038	0.0692	0.0629	0.0019	0.0610	0.0538	0.0030	0.0508
2	56	0.0860	0.0040	0.0820	0.0744	0.0020	0.0724	0.0641	0.0033	0.0608
3	48	0.0990	0.0044	0.0946	0.0855	0.0022	0.0833	0.0754	0.0037	0.0697
4	40	0.1120	0.0048	0.1072	0.0958	0.0024	0.0934	0.0813	0.0042	0.0771
5	40	0.1250	0.0048	0.1202	0.1088	0.0024	0.1064	0.0943	0.0042	0.0901
6	32	0.1380	0.0054	0.1326	0.1177	0.0027	0.1150	0.0997	0.0050	0.0947
8	32	0.1640	0.0054	0.1586	0.1437	0.0027	0.1410	0.1257	0.0050	0.1207
10	24	0.1900	0.0066	0.1834	0.1629	0.0033	0.1596	0.1389	0.0063	0.1326
12	24	0.2160	0.0066	0.2094	0.1889	0.0033	0.1856	0.1649	0.0063	0.1586
1/4	20	0.2500	0.0072	0.2428	0.2175	0.0036	0.2139	0.1887	0.0073	0.1814
5/16	18	0.3125	0.0082	0.3043	0.2764	0.0041	0.2723	0.2443	0.0081	0.2362
3/8	16	0.3750	0.0090	0.3660	0.3344	0.0045	0.3299	0.2983	0.0090	0.2893
7/16	14	0.4375	0.0098	0.4277	0.3911	0.0049	0.3862	0.3499	0.0101	0.3398
1/2	13	0.5000	0.0104	0.4896	0.4500	0.0052	0.4448	0.4056	0.0107	0.3949
9/16	12	0.5625	0.0112	0.5513	0.5084	0.0056	0.5028	0.4603	0.0117	0.4486
5/8	11	0.6250	0.0118	0.6132	0.5660	0.0059	0.5601	0.5135	0.0125	0.5010
3/4	10	0.7500	0.0128	0.7372	0.6850	0.0064	0.6786	0.6273	0.0136	0.6137
7/8	9	0.8750	0.0140	0.8610	0.8028	0.0070	0.7958	0.7387	0.0150	0.7237
1	8	1.0000	0.0152	0.9848	0.9188	0.0076	0.9112	0.8466	0.0166	0.8300
1 1/8	7	1.1250	0.0170	1.1080	1.0322	0.0085	1.0287	0.9497	0.0188	0.9309
1 1/4	7	1.2500	0.0170	1.2330	1.1572	0.0085	1.1487	1.0747	0.0188	1.0559
1 1/2	6	1.5000	0.0202	1.4798	1.3917	0.0101	1.3816	1.2955	0.0219	1.2734
1 3/4	5	1.7500	0.0232	1.7268	1.6201	0.0116	1.6085	1.5046	0.0260	1.4786
2	4 1/2	2.0000	0.0254	1.9746	1.8557	0.0127	1.8430	1.7274	0.0288	1.6986
2 1/4	4 1/2	2.2500	0.0254	2.2246	2.1057	0.0127	2.0930	1.9774	0.0288	1.9486
2 1/2	4	2.5000	0.0280	2.4720	2.3376	0.0140	2.3236	2.1933	0.0321	2.1612
2 3/4	4	2.7500	0.0280	2.7220	2.5876	0.0140	2.5736	2.4433	0.0321	2.4112
3	4	3.0000	0.0280	2.9720	2.8376	0.0140	2.8236	2.6933	0.0321	2.6612

TABLE 4—CLASS IIA, FREE FIT, FOR NUTS OF THE COARSE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Min.[2]	Tolerance	Max.	Min.[1]	Tolerance[3]	Max.	Min.	Tolerance	Max.
1	64	0.0741	0.0031	0.0772	0.0629	0.0019	0.0648	0.0561	0.0017	0.0578
2	56	0.0873	0.0033	0.0906	0.0744	0.0020	0.0764	0.0667	0.0019	0.0686
3	48	0.1005	0.0037	0.1042	0.0855	0.0022	0.0877	0.0764	0.0023	0.0787
4	40	0.1138	0.0042	0.1180	0.0958	0.0024	0.0982	0.0849	0.0027	0.0876
5	40	0.1268	0.0042	0.1310	0.1088	0.0024	0.1112	0.0979	0.0027	0.1006
6	32	0.1403	0.0049	0.1452	0.1177	0.0027	0.1204	0.1042	0.0034	0.1076
8	32	0.1663	0.0049	0.1712	0.1437	0.0027	0.1464	0.1302	0.0034	0.1336
10	24	0.1930	0.0063	0.1993	0.1629	0.0033	0.1662	0.1449	0.0045	0.1494
12	24	0.2190	0.0063	0.2253	0.1889	0.0033	0.1922	0.1709	0.0045	0.1754
1/4	20	0.2536	0.0072	0.2608	0.2175	0.0036	0.2211	0.1959	0.0054	0.2013
5/16	18	0.3165	0.0081	0.3246	0.2764	0.0041	0.2805	0.2524	0.0060	0.2584
3/8	16	0.3795	0.0090	0.3885	0.3344	0.0045	0.3389	0.3073	0.0068	0.3141
7/16	14	0.4427	0.0100	0.4527	0.3911	0.0049	0.3960	0.3602	0.0077	0.3679
1/2	13	0.5056	0.0107	0.5163	0.4500	0.0052	0.4552	0.4167	0.0084	0.4251
9/16	12	0.5685	0.0116	0.5801	0.5084	0.0056	0.5140	0.4723	0.0090	0.4813
5/8	11	0.6316	0.0124	0.6440	0.5660	0.0059	0.5719	0.5266	0.0098	0.5364
3/4	10	0.7572	0.0136	0.7708	0.6850	0.0064	0.6914	0.6417	0.0109	0.6526
7/8	9	0.8830	0.0150	0.8980	0.8028	0.0070	0.8098	0.7547	0.0120	0.7667
1	8	1.0090	0.0166	1.0256	0.9188	0.0076	0.9264	0.8647	0.0135	0.8782
1 1/8	7	1.1353	0.0188	1.1541	1.0322	0.0085	1.0407	0.9704	0.0154	0.9858
1 1/4	7	1.2603	0.0188	1.2791	1.1572	0.0085	1.1657	1.0954	0.0154	1.1108
1 1/2	6	1.5120	0.0222	1.5342	1.3917	0.0101	1.4018	1.3196	0.0180	1.3376
1 3/4	5	1.7644	0.0261	1.7905	1.6201	0.0116	1.6317	1.5335	0.0216	1.5551
2	4 1/2	2.0160	0.0288	2.0448	1.8557	0.0127	1.8684	1.7594	0.0241	1.7835
2 1/4	4 1/2	2.2660	0.0288	2.2948	2.1057	0.0127	2.1184	2.0094	0.0241	2.0335
2 1/2	4	2.5180	0.0321	2.5501	2.3376	0.0140	2.3516	2.2294	0.0270	2.2564
2 3/4	4	2.7680	0.0321	2.8001	2.5876	0.0140	2.6016	2.4794	0.0270	2.5064
3	4	3.0180	0.0321	3.0501	2.8376	0.0140	2.8516	2.7294	0.0270	2.7564

See page 528 for footnotes.

TABLE 5—CLASS IIB, MEDIUM FIT, FOR SCREWS OF THE COARSE THREAD SERIES

Size	Threads Per Inch	Major Diameter, in.			Pitch Diameter, in.			Minor Diameter, in.		
		Max.[1]	Tolerance	Min.	Max.[1]	Tolerance[3]	Min.	Max.[2]	Tolerance	Min.
1	64	0.0730	0.0028	0.0702	0.0629	0.0014	0.0615	0.0538	0.0025	0.0513
2	56	0.0860	0.0030	0.0830	0.0744	0.0015	0.0729	0.0641	0.0028	0.0613
3	48	0.0990	0.0032	0.0958	0.0855	0.0016	0.0839	0.0734	0.0031	0.0703
4	40	0.1120	0.0034	0.1086	0.0958	0.0017	0.0941	0.0813	0.0035	0.0778
5	40	0.1250	0.0034	0.1216	0.1088	0.0017	0.1071	0.0943	0.0035	0.0908
6	32	0.1380	0.0038	0.1342	0.1177	0.0019	0.1158	0.0997	0.0042	0.0955
8	32	0.1640	0.0038	0.1602	0.1437	0.0019	0.1418	0.1257	0.0042	0.1215
10	24	0.1900	0.0048	0.1852	0.1629	0.0024	0.1605	0.1389	0.0054	0.1335
12	24	0.2160	0.0048	0.2112	0.1889	0.0024	0.1865	0.1649	0.0054	0.1595
1/4	20	0.2500	0.0052	0.2448	0.2175	0.0026	0.2149	0.1887	0.0063	0.1824
5/16	18	0.3125	0.0060	0.3065	0.2764	0.0030	0.2734	0.2443	0.0070	0.2373
3/8	16	0.3750	0.0064	0.3686	0.3344	0.0032	0.3312	0.2983	0.0076	0.2906
7/16	14	0.4375	0.0072	0.4303	0.3911	0.0036	0.3875	0.3499	0.0088	0.3411
1/2	13	0.5000	0.0074	0.4926	0.4500	0.0037	0.4463	0.4056	0.0092	0.3964
9/16	12	0.5625	0.0080	0.5545	0.5084	0.0040	0.5044	0.4603	0.0101	0.4502
5/8	11	0.6250	0.0084	0.6166	0.5660	0.0042	0.5618	0.5135	0.0108	0.5027
3/4	10	0.7500	0.0090	0.7410	0.6850	0.0045	0.6805	0.6273	0.0117	0.6156
7/8	9	0.8750	0.0098	0.8652	0.8028	0.0049	0.7979	0.7387	0.0129	0.7258
1	8	1.0000	0.0108	0.9892	0.9188	0.0054	0.9134	0.8466	0.0144	0.8322
1 1/8	7	1.1250	0.0118	1.1132	1.0322	0.0059	1.0263	0.9497	0.0162	0.9335
1 1/4	7	1.2500	0.0118	1.2382	1.1572	0.0059	1.1513	1.0747	0.0162	1.0585
1 1/2	6	1.5000	0.0142	1.4858	1.3917	0.0071	1.3846	1.2955	0.0191	1.2764
1 3/4	5	1.7500	0.0164	1.7336	1.6201	0.0082	1.6119	1.5056	0.0216	1.4820
2	4 1/2	2.0000	0.0178	1.9822	1.8557	0.0089	1.8468	1.7274	0.0250	1.7024
2 1/4	4 1/2	2.2500	0.0178	2.2322	2.1057	0.0089	2.0968	1.9774	0.0250	1.9524
2 1/2	4	2.5000	0.0194	2.4806	2.3376	0.0097	2.3279	2.1933	0.0278	2.1655
2 3/4	4	2.7500	0.0194	2.7306	2.5876	0.0097	2.5779	2.4433	0.0278	2.4155
3	4	3.0000	0.0194	2.9806	2.8376	0.0097	2.8279	2.6933	0.0278	2.6655

TABLE 6—CLASS IIB, MEDIUM FIT, FOR NUTS OF THE COARSE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Min.[2]	Tolerance	Max.	Min.[1]	Tolerance[3]	Max.	Min.	Tolerance	Max.
1	64	0.0741	0.0026	0.0767	0.0629	0.0014	0.0643	0.0561	0.0017	0.0578
2	56	0.0873	0.0028	0.0901	0.0744	0.0015	0.0759	0.0667	0.0019	0.0686
3	48	0.1005	0.0031	0.1036	0.0855	0.0016	0.0871	0.0764	0.0023	0.0787
4	40	0.1138	0.0035	0.1173	0.0958	0.0017	0.0975	0.0949	0.0027	0.0876
5	40	0.1268	0.0035	0.1303	0.1088	0.0017	0.1105	0.0979	0.0027	0.1006
6	32	0.1403	0.0041	0.1444	0.1177	0.0019	0.1196	0.1042	0.0034	0.1076
8	32	0.1663	0.0041	0.1704	0.1437	0.0019	0.1456	0.1302	0.0034	0.1336
10	24	0.1930	0.0054	0.1984	0.1629	0.0024	0.1653	0.1449	0.0045	0.1494
12	24	0.2190	0.0054	0.2244	0.1889	0.0024	0.1913	0.1709	0.0045	0.1754
1/4	20	0.2536	0.0062	0.2598	0.2175	0.0026	0.2201	0.1959	0.0054	0.2013
5/16	18	0.3165	0.0070	0.3235	0.2764	0.0030	0.2794	0.2524	0.0060	0.2584
3/8	16	0.3795	0.0077	0.3872	0.3344	0.0032	0.3376	0.3073	0.0068	0.3141
7/16	14	0.4427	0.0087	0.4514	0.3911	0.0036	0.3947	0.3602	0.0077	0.3679
1/2	13	0.5056	0.0092	0.5148	0.4500	0.0037	0.4537	0.4167	0.0084	0.4251
9/16	12	0.5685	0.0100	0.5785	0.5084	0.0040	0.5124	0.4723	0.0090	0.4813
5/8	11	0.6316	0.0107	0.6423	0.5660	0.0042	0.5702	0.5266	0.0098	0.5364
3/4	10	0.7572	0.0117	0.7689	0.6850	0.0045	0.6895	0.6417	0.0109	0.6526
7/8	9	0.8830	0.0129	0.8959	0.8028	0.0049	0.8077	0.7547	0.0120	0.7667
1	8	1.0090	0.0144	1.0234	0.9188	0.0054	0.9242	0.8647	0.0135	0.8782
1 1/8	7	1.1353	0.0162	1.1515	1.0322	0.0059	1.0381	0.9704	0.0154	0.9858
1 1/4	7	1.2603	0.0162	1.2765	1.1572	0.0059	1.1631	1.0954	0.0154	1.1108
1 1/2	6	1.5120	0.0192	1.5312	1.3917	0.0071	1.3988	1.3196	0.0180	1.3376
1 3/4	5	1.7644	0.0227	1.7871	1.6201	0.0082	1.6283	1.5335	0.0216	1.5551
2	4 1/2	2.0160	0.0250	2.0410	1.8557	0.0089	1.8646	1.7594	0.0241	1.7835
2 1/4	4 1/2	2.2660	0.0250	2.2910	2.1057	0.0089	2.1146	2.0094	0.0241	2.0335
2 1/2	4	2.5180	0.0278	2.5458	2.3376	0.0097	2.3473	2.2294	0.0270	2.2564
2 3/4	4	2.7680	0.0278	2.7958	2.5876	0.0097	2.5973	2.4794	0.0270	2.5064
3	4	3.0180	0.0278	3.0458	2.8376	0.0097	2.8473	2.7294	0.0270	2.7564

See page 528 for footnotes.

TABLE 7—CLASS IIA, FREE FIT, FOR SCREWS OF THE FINE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Max.[1]	Tolerance	Min.	Max.[1]	Tolerance[3]	Min.	Max[2]	Tolerance	Min.
0	80	0.0600	0.0034	0.0566	0.0519	0.0017	0.0502	0.0447	0.0026	0.0421
1	72	0.0730	0.0036	0.0694	0.0640	0.0018	0.0622	0.0560	0.0028	0.0532
2	64	0.0860	0.0038	0.0822	0.0759	0.0019	0.0740	0.0668	0.0030	0.0638
3	56	0.0990	0.0040	0.0950	0.0874	0.0020	0.0854	0.0771	0.0033	0.0738
4	48	0.1120	0.0044	0.1076	0.0985	0.0022	0.0963	0.0864	0.0037	0.0827
5	44	0.1250	0.0046	0.1204	0.1102	0.0023	0.1079	0.0971	0.0039	0.0932
6	40	0.1380	0.0048	0.1332	0.1218	0.0024	0.1194	0.1073	0.0042	0.1031
8	36	0.1640	0.0050	0.1590	0.1460	0.0025	0.1435	0.1299	0.0045	0.1254
10	32	0.1900	0.0054	0.1846	0.1697	0.0027	0.1670	0.1517	0.0050	0.1467
12	28	0.2160	0.0062	0.2098	0.1928	0.0031	0.1897	0.1722	0.0057	0.1665
1/4	28	0.2500	0.0062	0.2438	0.2268	0.0031	0.2237	0.2062	0.0057	0.2005
5/16	24	0.3125	0.0066	0.3059	0.2854	0.0033	0.2821	0.2614	0.0063	0.2551
3/8	24	0.3750	0.0066	0.3684	0.3479	0.0033	0.3446	0.3239	0.0063	0.3176
7/16	20	0.4375	0.0072	0.4303	0.4050	0.0036	0.4014	0.3762	0.0073	0.3689
1/2	20	0.5000	0.0072	0.4928	0.4675	0.0036	0.4639	0.4387	0.0073	0.4314
9/16	18	0.5625	0.0082	0.5543	0.5264	0.0041	0.5243	0.4943	0.0081	0.4862
5/8	18	0.6250	0.0082	0.6168	0.5889	0.0041	0.5848	0.5568	0.0081	0.5487
3/4	16	0.7500	0.0090	0.7410	0.7094	0.0045	0.7049	0.6733	0.0090	0.6643
7/8	14	0.8750	0.0098	0.8652	0.8286	0.0049	0.8237	0.7874	0.0101	0.7733
1	14	1.0000	0.0098	0.9902	0.9536	0.0049	0.9487	0.9124	0.0101	0.9023
1 1/8	12	1.1250	0.0112	1.1138	1.0709	0.0056	1.0653	1.0228	0.0117	1.0111
1 1/4	12	1.2500	0.0112	1.2388	1.1959	0.0056	1.1903	1.1478	0.0117	1.1361
1 1/2	12	1.5000	0.0112	1.4888	1.4459	0.0056	1.4403	1.3978	0.0117	1.3861

[1] Basic diameters.
[2] Dimensions given are figured to the intersection of the worn tool arc with a centerline through crest and root.
[3] The tolerances specified for the pitch diameter are cumulative and include all errors of lead and angle.

TABLE 8—CLASS IIA, FREE FIT, FOR NUTS OF THE FINE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Min.[2]	Tolerance	Max.	Min.[1]	Tolerance[3]	Max.	Min.	Tolerance	Max.
0	80	0.0609	0.0024	0.0635	0.0519	0.0017	0.0536	0.0465	0.0013	0.0478
1	72	0.0740	0.0028	0.0768	0.0640	0.0018	0.0658	0.0580	0.0015	0.0595
2	64	0.0871	0.0031	0.0902	0.0759	0.0019	0.0778	0.0691	0.0017	0.0708
3	56	0.1003	0.0033	0.1036	0.0874	0.0020	0.0894	0.0797	0.0019	0.0816
4	48	0.1135	0.0037	0.1172	0.0985	0.0022	0.1007	0.0894	0.0023	0.0917
5	44	0.1266	0.0040	0.1306	0.1102	0.0023	0.1125	0.1004	0.0025	0.1029
6	40	0.1398	0.0042	0.1440	0.1218	0.0024	0.1242	0.1109	0.0027	0.1136
8	36	0.1660	0.0045	0.1705	0.1460	0.0025	0.1485	0.1339	0.0030	0.1369
10	32	0.1923	0.0049	0.1972	0.1697	0.0027	0.1724	0.1562	0.0034	0.1596
12	28	0.2186	0.0057	0.2243	0.1928	0.0031	0.1959	0.1773	0.0039	0.1812
1/4	28	0.2526	0.0057	0.2583	0.2268	0.0031	0.2299	0.2113	0.0039	0.2152
5/16	24	0.3155	0.0063	0.3218	0.2854	0.0033	0.2887	0.2674	0.0045	0.2719
3/8	24	0.3780	0.0063	0.3843	0.3479	0.0033	0.3512	0.3299	0.0045	0.3344
7/16	20	0.4411	0.0072	0.4483	0.4050	0.0036	0.4086	0.3834	0.0054	0.3888
1/2	20	0.5036	0.0072	0.5108	0.4675	0.0036	0.4711	0.4459	0.0054	0.4513
9/16	18	0.5665	0.0081	0.5746	0.5264	0.0041	0.5305	0.5024	0.0060	0.5084
5/8	18	0.6290	0.0081	0.6371	0.5889	0.0041	0.5930	0.5649	0.0060	0.5709
3/4	16	0.7545	0.0090	0.7635	0.7094	0.0045	0.7139	0.6823	0.0068	0.6891
7/8	14	0.8802	0.0100	0.8902	0.8286	0.0049	0.8335	0.7977	0.0077	0.8054
1	14	1.0052	0.0100	1.0152	0.9536	0.0049	0.9585	0.9227	0.0077	0.9304
1 1/8	12	1.1310	0.0116	1.1426	1.0709	0.0056	1.0765	1.0348	0.0090	1.0438
1 1/4	12	1.2560	0.0116	1.2676	1.1959	0.0056	1.2015	1.1598	0.0090	1.1688
1 1/2	12	1.5060	0.0116	1.5176	1.4453	0.0056	1.4515	1.4098	0.0090	1.4188

[1] Basic diameters.
[2] Dimensions given are figured to the intersection of the worn tool arc with a centerline through crest and root.
[3] The tolerances specified for the pitch diameter are cumulative and include all errors of lead and angle.

TABLE 9—CLASS IIB, MEDIUM FIT, FOR SCREWS OF THE FINE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Max.[1]	Tolerance	Min.	Max.[1]	Tolerance[3]	Min.	Max.[2]	Tolerance	Min.
0	80	0.0600	0.0026	0.0574	0.0519	0.0013	0.0506	0.0447	0.0022	0.0425
1	72	0.0730	0.0026	0.0704	0.0640	0.0013	0.0627	0.0560	0.0023	0.0537
2	64	0.0860	0.0028	0.0832	0.0759	0.0014	0.0745	0.0668	0.0025	0.0643
3	56	0.0990	0.0030	0.0960	0.0874	0.0015	0.0859	0.0771	0.0028	0.0743
4	48	0.1120	0.0032	0.1088	0.0985	0.0016	0.0969	0.0864	0.0031	0.0833
5	44	0.1250	0.0032	0.1218	0.1102	0.0016	0.1086	0.0971	0.0032	0.0939
6	40	0.1380	0.0034	0.1346	0.1218	0.0017	0.1201	0.1073	0.0035	0.1038
8	36	0.1640	0.0036	0.1604	0.1460	0.0018	0.1442	0.1299	0.0038	0.1261
10	32	0.1900	0.0038	0.1862	0.1697	0.0019	0.1678	0.1517	0.0042	0.1475
12	28	0.2160	0.0044	0.2116	0.1928	0.0022	0.1906	0.1722	0.0048	0.1674
1/4	28	0.2500	0.0044	0.2456	0.2268	0.0022	0.2246	0.2062	0.0048	0.2014
5/16	24	0.3125	0.0048	0.3077	0.2854	0.0024	0.2830	0.2614	0.0054	0.2560
3/8	24	0.3750	0.0048	0.3702	0.3479	0.0024	0.3455	0.3239	0.0054	0.3185
7/16	20	0.4375	0.0052	0.4323	0.4050	0.0026	0.4024	0.3762	0.0063	0.3699
1/2	20	0.5000	0.0052	0.4948	0.4675	0.0026	0.4649	0.4387	0.0063	0.4324
9/16	18	0.5625	0.0060	0.5565	0.5264	0.0030	0.5234	0.4943	0.0070	0.4873
5/8	18	0.6250	0.0060	0.6190	0.5889	0.0030	0.5859	0.5568	0.0070	0.5498
3/4	16	0.7500	0.0064	0.7436	0.7094	0.0032	0.7062	0.6733	0.0077	0.6656
7/8	14	0.8750	0.0072	0.8678	0.8286	0.0036	0.8250	0.7874	0.0088	0.7786
1	14	1.0000	0.0072	0.9928	0.9536	0.0036	0.9500	0.9124	0.0088	0.9036
1 1/8	12	1.1250	0.0080	1.1170	1.0709	0.0040	1.0669	1.0228	0.0101	1.0127
1 1/4	12	1.2500	0.0080	1.2420	1.1959	0.0040	1.1919	1.1478	0.0101	1.1377
1 1/2	12	1.5000	0.0080	1.4920	1.4459	0.0040	1.4419	1.3978	0.0101	1.3877

[1] Basic diameters.
[2] Dimensions given are figured to the intersection of the worn tool arc with a centerline through crest and root.
[3] The tolerances specified for the pitch diameter are cumulative and include all errors of lead and angle.

TABLE 10—CLASS IIB, MEDIUM FIT, FOR NUTS OF THE FINE THREAD SERIES

Size	Threads Per Inch	Major Diameter			Pitch Diameter			Minor Diameter		
		Min.[2]	Tolerance	Max.	Min.[1]	Tolerance[3]	Max.	Min.	Tolerance	Max.
0	80	0.0609	0.0022	0.0631	0.0519	0.0013	0.0532	0.0465	0.0013	0.0478
1	72	0.0740	0.0023	0.0763	0.0640	0.0013	0.0653	0.0580	0.0015	0.0595
2	64	0.0871	0.0026	0.0897	0.0759	0.0014	0.0773	0.0691	0.0017	0.0708
3	56	0.1003	0.0028	0.1031	0.0874	0.0015	0.0889	0.0797	0.0019	0.0816
4	48	0.1135	0.0031	0.1166	0.0985	0.0016	0.1001	0.0894	0.0023	0.0917
5	44	0.1266	0.0033	0.1299	0.1102	0.0016	0.1118	0.1004	0.0025	0.1029
6	40	0.1398	0.0035	0.1433	0.1218	0.0017	0.1235	0.1109	0.0027	0.1136
8	36	0.1660	0.0038	0.1698	0.1460	0.0018	0.1478	0.1339	0.0030	0.1369
10	32	0.1923	0.0041	0.1964	0.1697	0.0019	0.1716	0.1562	0.0034	0.1596
12	28	0.2186	0.0048	0.2234	0.1928	0.0022	0.1950	0.1773	0.0039	0.1812
1/4	28	0.2526	0.0048	0.2574	0.2268	0.0022	0.2290	0.2113	0.0039	0.2152
5/16	24	0.3155	0.0054	0.3209	0.2854	0.0024	0.2878	0.2674	0.0045	0.2719
3/8	24	0.3780	0.0054	0.3834	0.3479	0.0024	0.3503	0.3299	0.0045	0.3344
7/16	20	0.4411	0.0062	0.4473	0.4050	0.0026	0.4076	0.3834	0.0054	0.3888
1/2	20	0.5036	0.0062	0.5098	0.4675	0.0026	0.4701	0.4459	0.0054	0.4513
9/16	18	0.5665	0.0070	0.5735	0.5264	0.0030	0.5294	0.5024	0.0060	0.5084
5/8	18	0.6290	0.0070	0.6360	0.5889	0.0030	0.5919	0.5649	0.0060	0.5709
3/4	16	0.7545	0.0077	0.7622	0.7094	0.0032	0.7126	0.6823	0.0068	0.6891
7/8	14	0.8802	0.0087	0.8889	0.8286	0.0036	0.8322	0.7977	0.0077	0.8054
1	14	1.0052	0.0087	1.0139	0.9536	0.0036	0.9572	0.9227	0.0077	0.9304
1 1/8	12	1.1310	0.0100	1.1410	1.0709	0.0040	1.0749	1.0348	0.0090	1.0438
1 1/4	12	1.2560	0.0100	1.2660	1.1959	0.0040	1.1999	1.1598	0.0090	1.1688
1 1/2	12	1.5060	0.0100	1.5160	1.4459	0.0040	1.4499	1.4098	0.0090	1.4188

[1] Basic diameters.
[2] Dimensions given are figured to the intersection of the worn tool arc with a centerline through crest and root.
[3] The tolerances specified for the pitch diameter are cumulative and include all errors of lead and angle.

pitch diameter of the maximum screw and the minimum nut shall be zero for all pitches and all diameters

(6) Tolerance Values. The tolerances for a screw or nut of a given pitch shall be as specified in Table 2.

Screws, Bolts and Nuts

(Proposed Revision of S. A. E. Standard)

The Society has received inquiries from time to time as to standard practice for measuring lengths of screws and bolts, materials, heat-treatments and machining of the present S.A.E. Standard Screws, Bolts and Nuts shown on page C2 of the S.A.E. Handbook. These inquiries were referred to the Screw Threads Division for the purpose of revising the standard to include such data.

The Screw Threads Division recommends that the explanatory text in the S.A.E. Standard for Screws, Bolts and Nuts, page C2 of the S.A.E. Handbook, be revised and extended to read

The length of the effective thread of screws and bolts shall be: $D \times 1.5 + \frac{1}{4}$ in. As bolts and screws conforming to these specifications are primarily intended for use with nuts, the oval end is not included in the nominal length

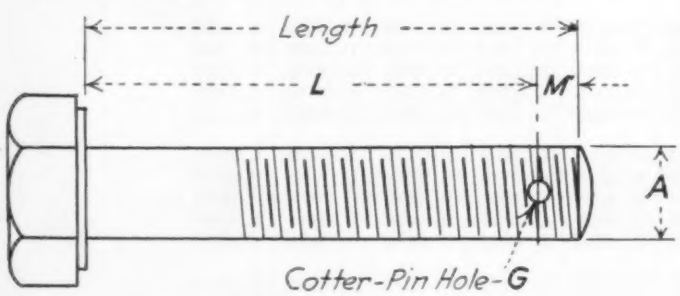
All heads and nuts shall be semi-finished. All screws and nuts shall be made of steel

Unless otherwise specified, S.A.E. Standard screws, bolts and nuts will be made to Class II A (Free Fits) as specified in the (proposed) S.A.E. Standard for Screw-Threads*

S.A.E. Standard bolts without slots or cotter-pin holes are obtainable in stock. If slots, cotter-pin holes or case-hardening are desired, these should be specified by the purchaser

The members of the Division have felt for some time that the present standard would be of more value if it included standard locations of the cotter-pin holes when for use where the latter are used. This subject was referred to Chairman Ehrman, of the Chicago Screw Co., who prepared a report which has been balloted upon favorably by the members of the Division. Therefore

The Screw-Threads Division recommends that the accompanying table indicating cotter-pin locations be added to the present S.A.E. Standard of Screws, Bolts and Nuts, page C2 of the S.A.E. Handbook.



Cotter-pin holes shall be located by dimension *L*

A†	*G*†	*M*	*A*†	*G*†	*M*	*A*†	*G*†	*M*
1/4–28	5/64	3/32	9/16–18	5/32	9/64	1 –14	5/32	3/16
5/16–24	5/64	3/32	5/8–18	5/32	5/32	1 1/8–12	7/32	7/32
3/8–24	1/8	7/64	11/16–16	5/32	5/32	1 1/4–12	7/32	7/32
7/16–20	1/8	7/64	3/4–16	5/32	3/16	1 3/8–12	1/4	1/4
1/2–20	5/32	1/8	7/8–16	5/32	3/16	1 1/2–12	1/4	1/4

*This proposed standard is taken from the Progress Report of the National Screw Thread Commission as reviewed by the Working Committee of the Sectional Committee, which is sponsored by the Society and the American Society of Mechanical Engineers.

†S.A.E. Standard, page C2, S.A.E. Handbook.

Gages and Gaging

(Proposed General Information)

The Screw Threads Division has undertaken to prepare articles covering various phases of screw-thread practice and matters germane thereto for publication as general information in The Journal and subsequently in the S.A.E. Handbook.

At the meeting of the Division held on May 1 Earl Buckingham, of Pratt and Whitney Company, who had been delegated to prepare a treatise on the fundamentals of gages and gaging for screw-thread products, submitted a report which was approved by the Division as general information.

The Division recommends that the following statement submitted by Mr. Buckingham be approved by the Standards Committee for publication as general information only.

GAGES AND GAGING FOR SCREW-THREADS

I. Introductory

The art of measuring screw-threads has developed very rapidly during the past few years. This development still continues, so that it would be extremely inadvisable to attempt to specify any one definite method as standard for this purpose. The object of this report is to establish so far as possible the fundamentals of this subject, and to point out various practices now successfully used.

II. Fundamentals

Object of Gaging.—The final result sought by gaging is interchangeable manufacture in some degree. This means that the mating parts can be assembled without *fitting* one part to another and, when assembled, the mechanism will function properly. Gaging should be employed more to prevent unsatisfactory parts from being produced than to sort out the correct parts from the incorrect ones.

Direction of Tolerances on Gages.—The extreme sizes for all limit-gages shall never exceed the extreme limits of the part being produced. All variations in the gages, whatever their cause or purpose, shall bring these gages within these extreme limits. Thus a gage that represents a minimum limit may be larger, but never smaller, than the minimum size specified for the part being produced, while the gage that represents a maximum limit may be smaller, but never larger, than the maximum size specified for the part being produced.

Temperature at Which Gages Shall Be Standard.—Gages shall be standard at a temperature of 68 deg. fahr.

Standard or Basic Size.—The standard or basic size, as physically represented by a correct standard master-gage, is the line at which interference begins between mating parts.

Purpose of "Go" and "Not Go" Gages.—The "Go" gages, which are the gages that represent the maximum limit of the internal member and the minimum limit of the external member, control the allowances between mating surfaces and also control interchangeability. "Go" gages control the maximum tightness in the fit of mating parts. Parts that are acceptable to proper "Go" gages will always interchange. Successful interchangeable manufacturing has been carried on for many years with the use of "Go" gages only.

The "Not Go" gages limit the extent of the permissible variations, thus limiting the amount of looseness between mating parts. "Not Go" gages control the maximum looseness in the fit of mating parts and thus control, in large measure, the proper functioning of the assembled mechanisms.

III. Gage Classification

Master Gage.—The master gage is a plug thread-gage that represents as exactly as possible the physical dimensions of the nominal or basic size of the component. A *standard master-gage* shall be accompanied by a record of its measurement and the gage should be used with knowledge of any deviations or corrections. In case of question, the deviations of this gage from the exact standard shall be ascertained by the Bureau of Standards.

Reference Gage.—A commonly used name for a master gage. Sometimes such gages include those that represent the extreme limits of the product and are used to check the inspection and working gages.

Gages Used to Measure the Product.—The gages used to check the product may be divided into two general types: mechanical and optical. Both types, however, are controlled by the master gages. In general, most of the parts accepted by one method of gaging will be accepted by the other. It should be pointed out, however, that those parts which are close in size to either rejection-point, may be accepted by one system and rejected by the other.

Mechanical gages are often divided into two classes: inspection gages and working gages. Inspection gages are for the use of the inspector in accepting the product. They are generally of the same design as the working gages and the dimensions are such that they represent very nearly the extreme limits of the part being produced. Working gages are those used by the workman to check the parts as they are machined. It is recommended that, when successive inspections are required, the working gages, by either design or selection, be of such dimensions that they are inside the limits of the gages used in succeeding inspections.

When gages of the optical type are employed, the same or duplicate instruments are used for both classes of inspection. No distinction in size is necessary, as the elements of wear and "feel" are not involved in this method of measuring.

IV. Gaging Practices and Gages

The production of accurate parts is primarily a matter of eternal vigilance. The smaller the limits that are to be maintained, the more complete the inspection or gaging system must be. To secure satisfactory results, the manufacturing tools provided must be sufficently accurate and the manufacturing methods sufficiently reliable to produce the required results. After tools and methods of proved reliability are provided, the next point is to watch the wear on the tools or their set-up to assure the maintenance of the required conditions. This is accomplished sometimes by a periodical test of the tools, sometimes by periodical gaging of the product, and sometimes by both.

Thread Micrometers.—Thread micrometers are used extensively to measure the pitch-diameter of taps and threaded internal parts. Thread micrometers should be calibrated periodically against a master gage, to avoid errors due to wear on the anvils of the instrument. Thread micrometers give no indication of lead and angle errors; therefore, the results of tests with thread micrometers alone cannot be taken as conclusive.

Thread Snap-Gages.—Thread snap-gages, generally consisting of conical points, are commonly used to measure the pitch-diameter of screws and other threaded internal parts. As in the case of thread micrometers, these gages give no indication of lead and angle errors. Therefore, the results of tests with them alone cannot be taken as conclusive.

Ring Thread-Gages.—Ring thread-gages are used extensively to measure the thread on internal parts. These are usually adjustable and are adjusted to suitable master or reference gages. Where parts are to be produced within specified limits, "Go" and "Not Go" gages are required. The thread on the "Go" gage is made of full form with its major diameter cleared or undercut to give a suitable clearance for grinding or lapping. The "Not Go" gage should be made primarily to check the minimum pitch-diameter. The minor diameter of such a gage should therefore never be smaller than the minor diameter of its corresponding "Go" gage, and its major diameter should be cleared as in the case of the "Go" gage. The use of such gages gives a certain measure of lead and angle errors, as well as of pitch-diameter errors. A proper "Go" gage will reject any parts that exceed the maximum dimensions specified. The "Not Go" gage, however, does not necessarily reject all parts that exceed the specified cumulative tolerance. It is possible, with the use of such gages, to accept parts that exceed this cumulative tolerance because of lead or angle errors, or both. With the proper check on tools and manufacturing methods, however, such possibilities are the exception. Such gages have been used successfully for many years.

Thread Comparators.—A recent development in the art of measuring threaded parts is the thread comparator, usually an optical instrument. These optical instruments throw an enlarged image of the thread upon a screen where it is compared with the enlarged outline of the required form. The location of the form used for comparison is made to agree with the image of the master gage. With such instruments all errors, both individual and cumulative, of diameter, lead and angle can be determined readily. These instruments can be adapted to measure taps and threaded internal parts.

Plug Thread-Gages.—Plug thread-gages are used exclusively at the present time to measure threaded holes or threaded external parts. Where parts are to be produced within specified limits, "Go" and "Not Go" gages are required. The thread on the "Go" gage is made of full form with its minor diameter cleared or undercut to give a suitable clearance for grinding or lapping. The "Not Go" gage should be made primarily to check the maximum pitch-diameter. The major diameter of such a gage should therefore never be larger than the major diameter of its corresponding "Go" gage, and its minor diameter should be cleared as in the case of the "Go" gage. The use of such gages gives a certain measure of lead and angle errors, as well as of pitch-diameter errors. A proper "Go" gage will reject any parts that exceed the minimum dimensions specified. The "Not Go" gage, however, as in the case of the ring thread-gage, does not necessarily reject all parts that exceed the specified cumulative tolerance.

Methods of Inspecting Screws.—One practice of inspecting screws produced on automatic machines is to provide a ring thread-gage set to approximately the mean size between the maximum and minimum limits. The threading tools are then set so that the product enters this intermediate gage, but will not enter the minimum or "Not Go" gage. The machine is then started up and parts are tested periodically with the regular "Go" gage and the intermediate gage. When the parts have increased in size so that they will not enter the intermediate gage more than three or four turns, the set-up is changed, even though the parts are still acceptable to the "Go" gage.

A very similar plan can be followed when a screw-thread comparator is employed. The original set-up should be toward the minimum limit and the set-up should be changed as the maximum limit is approached.

Reference has been made to successive inspections. Although the manufacturer may give but one inspection, it should be realized that the purchaser often inspects the product to assure that the prescribed specifications have been fulfilled. Therefore, to reduce the possibilities of disagreement to a minimum, the manufacturer should strive to produce parts well within the specified limits rather than close to the limiting sizes.

Thread micrometers and thread snap-gages are used extensively for testing the product as it is produced. As these instruments do not test all elements of the screw-thread, a "Go" gage should always be used as a supplementary test. Thread micrometers are a very effective means of watching the change in set-up due to wear on tools, etc.

Methods of Inspecting Tapped Holes.—One practice of inspecting tapped holes is first to inspect the tap, and then test the tapped holes periodically with suitable gages. The tap can be watched for wear by testing the tapped holes with a "Go" thread-gage. One widely used practice consists of using a "Go" plug thread-gage and a "Not Go" plain plug-gage for the minor diameter.

Another practice of inspecting taps is to measure the several elements, such as pitch-diameter, angle and lead; and still another consists of tapping a hole with each tap before it is issued from the tool-crib and testing these tapped holes with "Go" and "Not Go" plug thread-gages.

V. Inspection of Gages

When successive inspections in the same plant are involved, it is good practice to inspect all gages of the same nominal

size against each other periodically, and to distribute these gages so that the earlier inspections will be made with those that are the greatest amount inside of the component tolerance, and the later inspections with those gages closest in size to the component tolerance.

PASSENGER-CAR BODY DIVISION REPORT

Division Personnel

G. E. Goddard, *Chairman*	Dodge Bros.
A. J. Neerken, *Vice-Chairman*	Hupp Motor Car Corporation
Wm. Brewster	Brewster & Co.
E. G. Budd	Edward G. Budd Mfg. Co.
J. S. Burdick	Buffalo Body Corporation
O. H. Clark	Zeder-Skelton-Breer Engineering Co.
A. E. Garrels	Studebaker Corporation of America
E. W. Goodwin	Body Expert
G. W. Kerr	Rolls-Royce of America, Inc.
G. J. Mercer	Consulting Engineer
H. C. Nelson	Mullins Body Corporation

TOP-IRONS

(Proposed S.A.E. Recommended Practice)

One of the first subjects considered by the Passenger-Car Body Division was the standardization of top-irons. Information on present practice was obtained which indicated that a threaded-end with 7/16-14 U. S. Standard threads is used generally. Consideration was given to the possibility of standardizing a series of complete top-irons, and O. H. Clark, formerly with the Willys Corporation, was appointed a Subdivision to study the matter. A tentative recommendation was submitted which was printed in THE JOURNAL. Further consideration of the subject, however, indicated that only the threaded end of the top-iron should be standardized, as the other dimensions depends largely on individual conditions that do not lend themselves readily to interchangeability of top-irons. Therefore

> The Division recommends for S.A.E. Recommended Practice that the threaded-end of top-irons shall be 7/16-14 U. S. Standard thread

SPRINGS DIVISION REPORT

Division Personnel

S. P. Hess, *Chairman*	Detroit Steel Products Co.
H. R. McMahon, *Vice-Chairman*	Standard Steel Spring Co.
R. S. Begg	Jordan Motor Car Co.
E. S. Corcoran	Kelly-Springfield Motor Truck Co.
H. E. Figgie	Perfection Spring Co.
W. M. Newkirk	William & Harvey Rowland, Inc.
Gustof Peterson	Electric Alloy Steel Co.
E. V. Rippingille	Watson Stabilator Co.
F. A. Whitten	General Motors Truck Co.

SPRING-EYE BUSHINGS

(Proposed Revision of S.A.E. Recommended Practice)

The Springs Division has reviewed the existing spring standards and believes that the present S.A.E. Recommended Practice for Spring-Eye Bushings and Bolt Tolerances, page H5 of the S.A.E. HANDBOOK, should specify the tolerances for the spring-eye bushings only, the allowable bolt tolerances being properly a matter for the bolt manufacturers to decide. Therefore

> The Springs Division recommends that the present S.A.E. Recommended Practice for Spring-Eye Bushing and Bolt Tolerances be revised by the elimination of the bolt tolerances, the title of the Recommended Practice to be changed to "Spring-Eye Bushings."

DEFINITIONS

(Proposed S.A.E. Standard)

The Springs Division, in reviewing the existing Spring standards, has felt that definitions of terms used in the spring industry would be of material value in preventing many misunderstandings due to different interpretations of such terms. A Subdivision has been at work redrafting "Leaf-Spring Specifications" and in connection with this work the members of the Division have prepared the following definitions which are recommended for adoption by the Society.

This report probably should be adopted at this time as a section of the present S.A.E. Standard for Leaf-Spring Nomenclature, page H1 of the S.A.E. HANDBOOK. However, in the further development of standardization of automotive springs, it may be desirable to make these definitions a part of some other section of the standards, or an individual standard. This can be arranged for after the adoption of the definitions by the Society, as the work of the Division progresses.

> The Springs Division recommends the adoption of the following definitions as S.A.E. Standard:
>
> DEFINITIONS
>
> *Deflection.*—The amount of travel of a spring under the application of a specified load, expressed as inches per pound.
>
> *Flexibility.*—The flexing characteristics of a spring, determined by the weight which will deflect it 1 in., expressed as pounds per inch. (See "Flexibility Test," page H19, S.A.E. HANDBOOK.)
>
> *Composite Springs.*—Springs made of leaves (or plates), one or more of which are of alloy steel and the rest of carbon steel.
>
> *Unsprung Weight.*—The total weight of all parts not supported on the springs, including axles, wheels, rims and tires complete.
>
> *Rated Load.*—The load which the vehicle manufacturer specifies shall be carried by the springs at a given spring-height.
>
> *Load Clearance.*—The maximum distance through which a spring can travel beyond its rated load position, before striking.
>
> *Load Height.*—The distance from a line through the center of the spring-eyes to the face of the axle-pad, when the spring is deflected to rated load.
>
> *Load Length.*—The distance between the centers of the spring-eyes when the spring is deflected to rated load.
>
> *Free Height.*—The distance from a line through the center of the spring-eyes to the face of the axle-pad, when the spring is in a free, unloaded position.

FRAME BRACKETS FOR SPRINGS

(Proposed Cancellation of S.A.E. Recommended Practice)

In reviewing the present S.A.E. Recommended Practice for Frame Brackets for Springs, page H6 of the S.A.E. HANDBOOK, the Springs Division found that it is neither complete nor in accord with present practice. It is thought that the standard is of little value because each car builder has varying conditions to meet and determines its practice accordingly. Therefore

> The Springs Division recommends that the present S.A.E. Recommended Practice for Frame Brackets for Springs, page H6 of the S.A.E. HANDBOOK, be cancelled.

LUBRICANTS DIVISION REPORT

Division Personnel

H. C. Mougey, *Chairman*	General Motors Research Corporation
W. E. Jominy, *Vice-Chairman*	Studebaker Corporation
Sydney Bevin	Tide Water Oil Co.
P. J. Dasey	Midwest Engine Co.
A. P. Eves	International Harvester Co.
W. H. Herschel	Bureau of Standards
K. G. Mackenzie	Texas Co.
W. E. Perdew	Union Petroleum Co.
H. G. Smith	Atlantic Refining Co.
J. W. Stack	Standard Oil Co.

CRANKCASE LUBRICATING OIL

(Progress Report Only)

One of the most important subjects before the Standards Committee is the establishing of definite specifications for different grades of engine crankcase oil used by the several groups of the automotive industry. The Lubricants Division was reorganized on an active basis in 1921 and work started on formulating practical specifications for crankcase oils, cup greases, transmission greases and other classes of lubricant. The work of the Federal Government, conducted by the Bureau of Mines in the Interdepartmental Committee on the Standardization of Petroleum Specifications, and the standard methods of testing adopted by the American Society for Testing Materials, have been considered carefully by the Division.

In November 1921 a questionnaire was sent to passenger-car, motor-truck, tractor, motorcycle, motorboat, transmission and lubricant manufacturers, together with the Division's tentative specifications for crankcase oils, except for airplanes; motorcycle oils and transmission oils, requesting information as to changes or additions which should be made.

After a preliminary discussion of the tentative specifications at meetings of some of the Divisions representing automotive groups, a joint meeting of the Lubricants Division and members of the Agricultural Power Equipment, the Truck, the Passenger-Car, the Stationary-Engine, the Isolated Electric-Lighting Plant and the Engine Divisions and representatives of large oil producing companies was held in April.

The specifications were discussed thoroughly and revised somewhat in substance, heavier grades being added to the list and the separate specifications for automobile and motorcycle oils combined. The preparation of specifications for transmission lubricants has been referred to a Subdivision consisting of W. E. Jominy, of the Studebaker Corporation of America, and H. G. Smith, of the Atlantic Refining Co.

The Division is presenting the revised tentative proposal at this time as a progress report only, for the purpose of securing discussion from Standards Committee members and guests. It is hoped that this discussion will develop information and suggestions that will be of value to the Division in its work, so that, when the specifications are finally recommended for adoption by the Society, the Division may be assured that the specifications are approved generally and conform with the best lubricating-oil practice.

General.—These specifications cover grades of petroleum oil for the lubrication of internal-combustion engines, except aircraft, and are not recommended for the lubrication of turbines.

Only refined petroleum oils without admixture of fatty oils, resins, soaps or other compounds not derived from crude petroleum will be considered.

Corrosion Test.—The following corrosion test shall not cause discoloration of copper strip. Place a clean piece of mechanically polished pure strip copper about ½ in. wide and 3 in. long, and 10 cc. of the oil to be tested, in a clean test-tube. Close the tube with a vented stopper and hold for 3 hr. at 210 deg. fahr. Rinse the copper strip with sulphur-free acetone and compare it with a similar strip of freshly polished copper

Precipitation Number.—The precipitation number shall not be greater than 0.5 when determined by the method described in the American Society for Testing Materials Tentative Standard Method of Test for Pre-

ENGINE CRANKCASE OILS

Tentative specifications, April 18, 1922

General.—These specifications cover grades of petroleum oil for the lubrication of internal-combustion engines, except aircraft, and are not recommended for the lubrication of turbines.

Only refined petroleum oils without admixture of fatty oils, resins, soaps or other compounds not derived from crude petroleum will be considered.

Specification No.*	Flash-Point, Min.	Fire-Point, Min.	Viscosity Saybolt Sec. 100 Deg. Fahr. Min.	Viscosity Saybolt Sec. 100 Deg. Fahr. Max.	Viscosity Saybolt Sec. 210 Deg. Fahr. Min.	Viscosity Saybolt Sec. 210 Deg. Fahr. Max.	Color (NPA) Darkest color allowed on mixture of oil and 50 per cent kerosene	Pour Test, Max.	Acidity, Mg. KOH per Gram., Max.	Conradson Carbon Residue, Per Cent, Max.	Precipitation Number
20	325	365	180	220	43	....	5	35	0.15	0.20	....
020	325	365	180	220	43	....	5	0	0.15	0.20	....
30	335	380	270	330	46	....	5	40	0.15	0.30	....
030	335	380	270	330	46	....	5	0	0.15	0.30	....
40	345	390	360	440	49	....	5	45	0.15	0.40	....
50	355	400	450	550	52	....	6	50	0.15	0.60	....
60	360	.....	.....	.....	55	65	....	55	0.15	0.80	....
80†	400†	.....	.....	.....	75	85	....	15	0.15	1.5	0.5
95	400	.....	.....	.....	90	100	....	45	0.15	1.5	0.5

A corrosion test is required for all grades.

*For Specifications Nos. 20 to 50 inclusive, the numbers indicate the first two figures of the average Saybolt viscosity in seconds at 100 deg. fahr. of the grades indicated. The cipher preceding Specifications Nos. 20 and 30 indicate that the pour-test value of these two grades is zero. Nos. 60, 80 and 95 indicate the average Saybolt viscosity in seconds for these three grades at 210 deg. fahr.

†The limits for No. 80 are to be reviewed by the Lubricants Division, as it is felt that the pour-test limit for this grade of oil is low.

cipitation Number of Lubricating Oils, No. D91-21T. The precipitation number is the number of cubic centimeters of precipitate formed when 10 cc. of lubricating oil is mixed with 90 cc. of petroleum naphtha of definite quality and centrifuged under definite prescribed conditions

OBITUARIES

FRANK W. EDWARDS, sales engineer of the Dayton Engineering Laboratories Co., died April 29 at his residence in Dayton, Ohio, aged 41 years. His death was due to an acute case of pneumonia. He was born at Georgetown, Ohio, April 10, 1881, and received his education in the high school and the night school of the Y. M. C. A.

In 1901 Mr. Edwards entered the service of the Central Union Telephone Co., as a stock keeper and before leaving in July, 1905, had been promoted to assistant wire chief. From the Central Union Telephone Co. he went to the National Cash Register Co. at Dayton, remaining there 7 years. He was first connected with the electrical department and after serving there for a while was later engaged in installing electrical systems in the department stores of that city. On Aug. 1, 1912, he became associated with the Dayton Engineering Laboratories Co., serving in its experimental laboratory for 6 months and then having charge of the installing of experimental starting, lighting and ignition systems at automobile factories for 2 years. From 1914 until the time of his death Mr. Edwards served in the sales department with the title of sales engineer.

Mr. Edwards was elected to Associate Member grade in the Society Sept. 8, 1915, and was transferred to Member grade June 16, 1918.

HENRY HESS, one of the founder members of the Society and president in 1909, died on March 23, 1922, at his home in Atlantic City, N. J. At the time of his death Mr. Hess was president and chief engineer of the Hess Steel Corporation which he organized in 1912.

Henry Hess was born in Darmstädt, Germany, in March, 1864. He received his early education in private schools at New York City, later studying in Germany. His practical experience was obtained in the machine shop, drawing room and office, in turn as machinist, foreman, draftsman and manager, chiefly with machine-tool builders and the United States Government. While in the service of the Niles Tool Works, he designed its German plant and was sent to Oberschoeneweide near Berlin where he had charge of its erection, remaining there several years as director and consulting engineer. On his return to the United States in 1902 he organized the Hess-Bright Mfg. Co. for the manufacture and sale of ball bearings conforming to the German D. W. and M. design. He sold his interests in this company in 1912, when he organized the Hess Steel Corporation of Baltimore, serving as president and chief engineer of this company until his death.

Mr. Hess was very active in the early standardization work of the Society that has become of such importance to the industry at large. He was chairman of the Gear-Tooth Shapes and the Technical Index Divisions from 1911 to 1914 and a member of the Ball and Roller Bearings Division from 1910 to 1913. He was also chairman of the Pennsylvania Section of the Society from 1911 to 1913. He was a member of the American Society of Mechanical Engineers, serving as manager from 1911 to 1914 and as vice-president from 1914 to 1916. He was also a past-president of the Philadelphia Engineers' Club, a member of the American Institute of Mining Engineers, the American Society for Testing Materials, the American Iron and Steel Institute, the American Electrochemical Society, the American Academy of Political and Social Science, the Franklin Institute, the New York Engineers' Club, the Art Club and the Economics Club.

Mr. Hess is survived by his widow, two daughters and a son, H. Lawrence Hess, a member of the Society.

TEST APPARATUS FOR PROPELLER MODELS

SPECIAL apparatus of interesting design has been devised by S. Albert Reed, New York City, for the testing of high-speed propeller models. In this apparatus an electric motor is employed to drive the propeller-shaft through a set of gears that increases the motor speed four times. This gear is balanced, that is, mounted on two countershafts, an arrangement which is designed to relieve the shaft of all strains except that of torsion inasmuch as the shaft is floating. Instead of providing for thrust play by a sliding joint in the shaft, the entire propeller-shaft with its pinion slides to and fro between the countershaft gears, thus doing away with any binding due to the fact that the gear teeth mesh and disengage at a high rate of speed.

The longitudinal play of the propeller-shaft was approximately ½ in. and the shaft has a collar with a ball-bearing against it that is carried by hinged lever, the free end of which is held back by a spring scale for measuring the thrust. The thrust bearing presented some novel problems inasmuch as a maximum thrust of 500 lb. at 6000 r.p.m. must be provided for, which is considerably in excess of the usual range of ball-bearing practice. This problem was solved by the use of tandem ball-bearings, in which the speed of each was reduced in the ratio of the number in tandem. To measure the torque the stress on the countershafts transmitting the motion from the motor shaft to the propeller-shaft was utilized by an extension of the principle of the well-known transmission dynamometer. The torque of the frame or box carrying the countershafts has a certain fixed ratio to the speed and horsepower being transmitted at any minute and this value was transmitted by an arm to a spring scale that gave accurate measurements.

FACILITY OF ASSOCIATION

THE spirit of suspicion, selfishness, intolerance and controversy complicates all our problems, domestic and international. Racial prejudices, together with narrow and mistaken views of national interests, are responsible for wars. We wonder at industrial disputes, in which questions of immediate personal interest are involved and where the participants are often lacking in education and breadth of culture.

The truth is that it is very hard for people to get along together without friction and antagonism. Emerson said that "facility of association" is the measure of civilization, meaning that as people become enlightened and broad-minded they develop the ability to cooperate in higher and higher degree. The ability to work together harmoniously and effectively for common interests and purposes is the measure of civilization and the general condition of social progress.

The gains of society in the last 100 years have been accomplished for the most part by improvements in the methods of production, by the use of power and machinery. They have been accomplished by the development of the industrial plant, and the industrial plant represents the earnings and savings and profits of individuals. If the profits had been less, the industrial development would have been less, and the evils of scarcity and high prices would have been greater than they are.

In the 10 years from 1899 to 1909, population in the United States increased 21 per cent, while the amount of capital invested in the manufacturing industries increased 105 per cent, and the amount of power employed in these industries increased 85 per cent.—George E. Roberts.

ACTIVITIES OF THE SECTIONS

Secretaries of the Sections

BUFFALO SECTION—A. J. Fitzgibbons, 168 Claremont Avenue, Buffalo
CLEVELAND SECTION—E. W. Weaver, 5103 Euclid Avenue, Cleveland
DAYTON SECTION—R. B. May, Dayton Engineering Laboratories, Dayton
DETROIT SECTION—Thomas J. Lítle, Jr., 733 Seyburn Avenue, Detroit
Mrs. B. Brede, Assistant Secretary, 1361 Book Building, Detroit
INDIANA SECTION—B. F. Kelly, Weidely Motors Co., Indianapolis
METROPOLITAN SECTION—R. E. Plimpton, 129 East 45th Street, New York City
MID-WEST SECTION—H. O. K. Meister, Hyatt Roller Bearing Co., 2715 South Michigan Avenue, Chicago
MINNEAPOLIS SECTION—Phil N. Overman, 10 South 10th Street, Minneapolis
NEW ENGLAND SECTION—V. A. Nielsen, 701 Beacon Street, Boston.
PENNSYLVANIA SECTION—T. F. Cullen, Chilton Co., Market and 49th Streets, Philadelphia
WASHINGTON SECTION—Benjamin R. Newcomb, 211 Victor Building, City of Washington

THE Sections of the Society closed the spring season of activitiy with their May meetings in most cases. One or two of the Sections are planning summer outings of a social nature but in general they will hold no meetings until the fall. The new Section officers took up the reins during the month and will be responsible for the programs arranged for the next Section year. The interest in Section meetings this past winter has been gratifying, but it is the ambition of the Council and the Sections Committee to encourage even greater activity during the coming year. With this in mind, a special Sections luncheon will be a feature of the White Sulphur Springs Meeting. Representatives of all of the Sections will meet with the Councilors and the Sections Committee to discuss ways and means of attaining maximum interest in Section meetings and planning them to render the greatest service to the industry on engineering problems that are urgent. Several of the members who have been especially active and successful in the administration of Sections affairs will address the luncheon and relate their experiences for the benefit of the new Section officers.

NEW SECTION OFFICERS

The result of the election of new officers was announced by all of the Sections at their May meetings except in the case of the Pennsylvania Section. Study of the following list convinces one that considerable care was taken by the respective nominating committees in the making of selections, and forecasts an active year among the Section members.

BUFFALO SECTION

E. O. Spillman	Chairman
C. A. Criqui	Vice-Chairman
Otto M. Burkhardt	Treasurer
A. J. Fitzgibbins	Secretary

CLEVELAND SECTION

O. A. Parker	Chairman
R. J. Nightingale	Vice-Chairman
H. E. Figgie	Treasurer
E. W. Weaver	Secretary

DAYTON SECTION

J. H. Hunt	Chairman
Iskander Hourwich	Vice-Chairman
Robert F. McCann	Treasurer
R. B. May	Secretary

DETROIT SECTION

George E. Goddard	Chairman
K. K. Hoagg	Vice-Chairman
Charles S. Whitney	Treasurer
Thomas J. Lítle, Jr.	Secretary

INDIANA SECTION

O. C. Berry	Chairman
Lon R. Smith	Vice-Chairman
Mark A. Smith	Treasurer
B. F. Kelly	Secretary

METROPOLITAN SECTION

W. E. Kemp	Chairman
H. W. Slauson	Vice-Chairman
W. P. Kennedy	Treasurer
R. E. Plimpton	Secretary

MID-WEST SECTION

Taliaferro Milton	Chairman
Benjamin S. Pfeiffer	Vice-Chairman
Nelson B. Nelson	Treasurer
H. O. K. Meister	Secretary

MINNEAPOLIS SECTION

A. H. Bates	Chairman
Victor Gauvreau	Vice-Chairman
J. S. Clapper	Treasurer
Phil Overman	Secretary

NEW ENGLAND SECTION

H. E. Morton	Chairman
H. F. Peavey	Vice-Chairman
L. B. F. Raycroft	Treasurer
V. A. Nielsen	Secretary

WASHINGTON SECTION

Col. F. H. Pope	Chairman
Chester H. Warrington	Vice-Chairman
Conrad H. Young	Treasurer
Benj. H. Newcomb	Secretary

INTER-SECTION ATHLETICS

One of the features of the sports program at the White Sulphur Springs Meeting will be the athletic struggle to determine which Section will be the Sports Champion of 1922. At Summer Meetings of the past, only two or three Inter-Section events have been arranged. The plan this year calls for the crediting of points to the Section represented by the winners, seconds and thirds in all of the numerous events. At the conclusion of the sports program an appropriate cup will be awarded to the Section that boasts the greatest total of points. The battle for supremacy and the good-natured rivalry between Sections will create an atmosphere not unlike that of the inter-collegiate athletic meets of our younger days. The baseball cup emblematic of the Section Championship will be contested between Section teams; Section running and swimming relay races are scheduled; and even the points scored by the ladies will be officially

recognized in the totals. Reports reaching the Society offices tell of committees that are busily organizing ball clubs and influencing golf, tennis and sprint stars to present themselves for action at White Sulphur Springs.

Section Stunts

A Summer Meeting would not be complete without the famous Section Stunts. Some of the brightest moments of the entertainment are to be provided by the near-actors, virtuosos or circus ringmasters who represent one of the local organizations of the Society. Surprises are being prepared for the June meeting and, of course, a dense cloud of secrecy has been thrown about the plans. The Mid-West Booster, official organ of the Chicago lads, carries in part the following for May, which we cannot refrain from reprinting.

Here are some ideas:

(1) That all of us go to White Sulphur without shoes or socks. Have all the rest of the clothes and be regular in every way; except go barefoot. This would attract some attention

(2) That all of us who go, whenever we are in the presence of people not members of the Mid-West Section, act like deaf-and-dummies and pretend only to be able to talk with our hands. This will save us lots of useless conversation with Eastern reubens. We can learn the finger language on the way down, which will leave us no time for ruinous games of chance

(3) That all of us grow a full beard and mustaches and every fellow call every other fellow in our party "Doc" and we will all wear green spectacles. These whiskers will give us some publicity and will add great dignity to our Section in the eyes of the Bostonese

(4) That every single one of us equip himself with working drawings and talking specifications of some great invention, only they won't be great inventions, only we will all pretend to believe in every one of them and talk publicly and privately about them every chance we get. One fellow can have a scheme for a rain-water substitute for gasoline; another can have a seven-cylinder engine, the extra cylinder being turned on when climbing a hill; another can have a rubber substitute made from the refuse of Italy's macaroni factories, and so on

If this is an indication of what may be expected from the Windy City, the other Sections will surely be put to shame.

Section Pullmans

The Cleveland, Detroit, Mid-West, Metropolitan and Pennsylvania Sections have all arranged to have special Pullman cars transport their members to the Summer Meeting. The Dayton Section members can be accommodated on either the Detroit or Cleveland cars. The Indiana members can make reservations through the Mid-West Section on their cars. The New England and the Washington Sections members can make reservations on the Eastern special train with the Metropolitan and the Pennsylvania Sections. The Section Secretaries have complete information on all of these special Pullmans. Consult the one nearest you and travel to White Sulphur Springs with your friends in the Society.

Mid-West Section

The new officers of the Mid-West Section devoted the meeting on May 12 to a social gathering for the purpose of discussing plans for the fall. Chairman Milton expressed a desire to schedule papers having the greatest general interest. Many good suggestions were offered.

Metropolitan Section

The Metropolitan Section's trip to New Haven via railway motor-cars on the occasion of its April Meeting aroused a surprising amount of interest. As a consequence, the Section officers decided to make Motor Cars on Rails the subject of discussion at the meeting on May 25. The meeting was opened with papers on motor rail-cars by Roy V. Wright and W. L. Bean. Following these, there was an active discussion of the engineering problems involved in the design, construction and operation of this type of automotive vehicles. The meeting was preceded by a well attended dinner.

Minneapolis Section

The Minneapolis Section held an informal meeting of a social character on the evening of May 3 at the Manufacturers Club. The annual reports of officers and committees were read and officers elected for the next Section year. L. A. Emerson was selected to represent the Section on the Society Nominating Committee at White Sulphur Springs.

Pennsylvania Section

The members of the Pennsylvania Section held a very enjoyable outing at the Torresdale Golf Club on May 26. Several heated golf and tennis contests were staged, the ladies being present to enjoy the fun. A palatable dinner was served after the athletic diversions in the afternoon.

Indiana Section

President Bachman and several members of the Council were guests of the Indiana Section at its dinner and social meeting on May 8. Mark A. Smith was elected representative on the Society Nominating Committee and W. G. Wall was selected as alternate. Short talks were given by President Bachman, Past-President Beecroft and others.

Washington Section

The Washington Section held a meeting on the evening of May 5 in the lecture hall of the Cosmos Club. Dr. Harvey Wiley was the speaker and chose as his topic the Metric System and its Practical Use. In view of present agitation in some circles to have the metric units declared by the Congress the national standard, Dr. Wiley's talk resulted in an active general discussion in which he was called upon to answer numerous questions. A buffet supper was provided for the members after the meeting adjourned.

Cleveland Section

Dr. H. C. Dickinson addressed the Cleveland Section members at their meeting on May 19, having as a topic the work being done by the Research Department of the Society. He explained some of the advantages of cooperative research on specific problems and indicated the scope of the Society's activity in this field. William T. Burns presented a paper on power losses in bearings which included valuable tabulated data. The meeting was held in the Hotel Winton and was preceded by an informal dinner.

New England Section

In addition to the election of Section officers, the New England Section elected its representatives on the Society Nominating Committee at the meeting on May 11. L. W. Rosenthal was selected as member and Prof. E. P. Warner was chosen as alternate. After enjoying an informal dinner, the Section heard an extremely interesting paper on the Making of Automobile Tires which was presented by W. W. Duncan. Mr. Duncan illustrated his paper with slides and gave a very clear description of the successive operations followed in building a pneumatic tire.

Detroit Section

The Detroit Section devoted its meeting on May 19 to papers on related subjects, Methods of Producing Cylinder-Wall Surfaces, and Light-Weight Pistons. A. J. Baker discussed present-day methods of machining cylinder-bores and H. B. Carlson made remarks on cylinder lapping. John A. Gann read a paper on light-weight pistons cast from special alloys, and E. Planche presented the results of tests he had made with cars whose engines were equipped with alloy pistons of this type. The meeting was very well attended and was preceded by a buffet supper.

Applicants *for* Membership

The applications for membership received between April 15 and May 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ANDERSON, KENNETH B., student, California Institute of Technology, *Pasadena, Cal.*

ATANGAN, TOMAS T., 3800 Grand Boulevard, *Chicago.*

BARNES, ORRIN HAYWARD, student, California Institute of Technology, *Pasadena, Cal.*

BEEGLE, F. N., president, Union Drawn Steel Co., *Beaver Falls, Pa.*

BELL, STANLEY A., student, California Institute of Technology, *Pasadena, Cal.*

BIDDLE, CHARLES JONATHAN, student, California Institute of Technology, *Pasadena, Cal.*

BLAKELY, LOREN E., student, California Institute of Technology, *Pasadena, Cal.*

BOWER, BYRON F., student, Ohio State University, *Columbus, Ohio.*

BRAINERD, HOWARD S., metallurgist, Ingersoll-Rand Co., *Phillipsburg, N. J.*

BRAMLEY, N. F., president and general manager, Templar Motors Co., *Cleveland.*

BREAKEY, EDWIN T., factory superintendent, Columbia Carbureter Co., *Chicago.*

BRICE, JOHN R., layout draftsman, Yellow Cab Mfg. Co., *Chicago.*

BROSSEAU, A. J., president, Mack Trucks, Inc., 25 Broadway, *New York City.*

BRUNNER, ALEXANDER, proprietor, A. Brunner & Son, *Newark, N. J.*

BUELL, ROY D., president, Buell Mfg. Co., *Chicago.*

COWAN, WILLIAM ARTHUR, assistant chief chemist, National Lead Co., *New York City.*

CREAMER, CHARLES DELL, student, Ohio State University, *Columbus, Ohio.*

DE BARRIN, HENRY, engineer designer, Sociedad Automoviles Argus S. A., *Madrid, Spain.*

DE MELTO, VINCENT MARION, student, Ohio State University, *Columbus, Ohio.*

DEMORY, A. R., president, Timken-Detroit Axle Co., *Detroit.*

DICK, JAMES B., production engineer, D'Arcy Spring Co., *Kalamazoo, Mich.*

DOTY, E. M., mechanical engineer, Denby Motor Truck Co., *Detroit.*

DROPINSKI, ADOLPH, student, Armour Institute of Technology, *Chicago.*

DRYER, JAMES C., vice-president, James Cunningham Son & Co., *Rochester, N. Y.*

DUPONT, V. H. MEYER, founder, publisher and sole owner, *Oriental Motor, Shanghai, China.*

EDMUNDS, GEORGE E., president, Edmunds & Jones Corporation, *Detroit.*

FETHERSTON, W. L., manager of trade division, Robert Bosch Magneto Co., *New York City.*

FITZGERALD, EDWARD WILLIAM, inspector of tanks and tractors, Ordnance Department, *Camp Meade, Md.*

FLOYD, ROBERT K., general manager, F. H. Floyd, *Detroit.*

FOGELSON, EMIL, mechanical engineer, Victor Pagé Motors Corporation, *New York City.*

FUHRER, MAX, student, Armour Institute of Technology, *Chicago.*

GIBSON, ALFRED R., 210 Walnut Street, *Montclair, N. J.*

GOLDSTEIN, ALEXANDER, student, Armour Institute of Technology, *Chicago.*

GOMMEL, D. E., chief engineer, Monroe Automobile Co., *Indianapolis.*

GRANNING, M. L., instructor, Oregon Agricultural College, *Corvallis, Ore.*

GRIFFITH, EDWARD W., factory representative, Forsyth Bros. Co., *Harvey, Ill.*

HATCH, DORRILL K., technical sales, Gurney Ball Bearing Co., *Jamestown, N. Y.*

HERBERT, W. H., general sales manager, Denby Motor Truck Co., *Detroit.*

HERRLIN, KARL O., student, Lewis Institute, *Chicago.*

HIGHAM, WILLIAM HARRY MARCUS, senior draftsman, Durant Motors, Inc., *Long Island City, N. Y.*

HOFFMAN, ROBERT J., works manager, Prest-O-Lite Co., Inc., *New York City.*

HOHNKE, JOHN H., student, Michigan Agricultural College, *East Lansing, Mich.*

HOLDER, H. A., president, R. & V. Motor Co., *East Moline, Ill.*

HOSLEY, LORING F., chief engineer, Kelly-Springfield Motor Truck Co., *Springfield, Ohio.*

HOUSE, BRYAN, designer, Maxwell Motor Co., Inc., *Detroit.*

HUEBOTTER, H. A., research assistant, Purdue University, *Lafayette, Ind.*

HULT, ALBERT E., assistant general manager, Detroit Motorbus Co., *Detroit.*

KNERR, HORACE C., metallurgist, Navy Yard, *Philadelphia.*

KNIGHT, RALPH B., superintendent of inspection, North East Electric Co., *Rochester, N. Y.*

KOHR, ROLAND MEREDITH, student, Ohio State University, *Columbus, Ohio.*

KRYZ, EMIL W., mechanical draftsman, Rock Island Arsenal, *Rock Island, Ill.*

LAWSON, CARL, inspector, J. M. Horton Ice Cream Co., *New York City.*

MCDONNELL, EDWARD O., vice-president and general manager, Kelly-Springfield Motor Truck Co., *Springfield, Ohio.*

MACDUFF, D. M., chief engineer, A. J. Detlaff Co., *Detroit.*

MAGDSICK, H. H., engineering department, National Lamp Works of General Electric Co., *Nela Park, Cleveland.*

MARSH, HALLAN N., student, California Institute of Technology, *Pasadena, Cal.*

MARSTON, R. E., executive engineer, Selden Truck Corporation, *Rochester, N. Y.*

MEADER, GLENN S., student, University of Minnesota, *Minneapolis.*

MIKI, KICHIHEI, Frazar & Co., 30 Church Street, *New York City.*

MILLER, H. S., general manager, Plowman Tractor Co., *Waterloo, Iowa.*

MORTON, ALLEN WELLER, chief engineer, American Hammered Piston-Ring Co., *Baltimore.*

MORRIS, VICTOR ROSS, student, Ohio State University, *Columbus, Ohio.*

PARISH & BINGHAM CORPORATION, *Cleveland.* (Affiliate Membership)

PARROTT, R. B., secretary and treasurer, Service Products Corporation, *Indianapolis.*

PETTEBONE, ORLANDO R., draftsman, C. L. Best Tractor Co., *San Leandro, Cal.*

PHILIP, CHARLES W., chief engineer, Philip Motor Co., *San Francisco.*

PRESTON, NORMAN A., used car mechanical superintendent, Detroit Cadillac Motor Car Co., *New York City.*

RADNER, SAMUEL, student, Armour Institute of Technology, *Chicago.*

RANDOLPH, CHARLES L. W., student, Lewis Institute, *Chicago.*

REEVES, HUBERT A., student, California Institute of Technology, *Pasadena, Cal.*

ROEMMELE, HOWARD CARL, student, Stevens Institute of Technology, *Hoboken, N. J.*

ROHLOFF, DEWEY CHARLES, student, California Institute of Technology, *Pasadena, Cal.*

ROOT, RALPH C., engineer and sales manager, Service Products Corporation, *Indianapolis.*

Schenck, R. B., metallurgical engineer, Buick Motor Co., *Flint, Mich.*

Schneider, Harold P., student, Ohio State University, *Columbus, Ohio.*

Schneider, Warren Arthur, student, California Institute of Technology, *Pasadena, Cal.*

Sebbes, Louis H., advertising copy writer, *Automotive Industries, Chicago.*

Sherrard, Joe O., student, Ohio State University, *Columbus, Ohio.*

Skriba, Louis S., student, Armour Institute of Technology, *Chicago.*

Small, F. M., president, Martin-Parry Corporation, *York, Pa.*

Smellie, Edwin Frost, student, University of Michigan, *Ann Arbor, Mich.*

Sower, George W., student, Ohio State University, *Columbus, Ohio.*

Steiner, Felix P., assistant tool designer, Detroit Cadillac Motor Car Co., *New York City.*

Stettinius, W. C., sales manager, American Hammered Piston-Ring Co., *Baltimore.*

Stevenson, Horace N., foreman, Mercer Motors, *Trenton, N. J.*

Stroh, Edwin R., vice-president, Stroh Casting Co., *Detroit.*

Tetens, Raymond E., assistant chief engineer, Earl Motors, Inc., *Jackson, Mich.*

Thompson, Chester A., general manager, Tire & Rim Association, *Cleveland.*

Trautman, Harry A., foreman, Steel Products Co., *Cleveland.*

Vaughan, Samuel E., student, Leland Stanford, Jr., University, *Stanford University, Cal.*

Verplank, A. J., student, Armour Institute of Technology, *Chicago.*

Vickers, William Harry, student, Armour Institute of Technology, *Chicago.*

Voorhees, R. C., testing engineer, R. F. D. No. 3, *Ypsilanti, Mich.*

Walworth, Richard H., student, Armour Institute of Technology, *Chicago.*

Wickel, Ralph O., student, Armour Institute of Technology, *Chicago.*

Wilson, Edward A., student, California Institute of Technology, *Pasadena, Cal.*

Woolson, Harry Thurber, engineer, Zeder Motor Car Co., *Newark, N. J.*

Wyzalek, John F., metallurgist, Hyatt bearings division, General Motors Corporation, *Harrison, N. J.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between April 10 and May 10, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

Bacon, Elbridge F. (E S) student, University of Michigan, *Ann Arbor, Mich.*, (mail) 1038 East Huron Avenue.

Caldwell, Jesse Thomas (A) commercial engineer, National Lamp Works of General Electric Co., *Nela Park, Cleveland.*

DeFrance, Smith J. (E S) student, University of Michigan, Ann Arbor, Mich., (mail) 27 Lansing Avenue, *Battle Creek, Mich.*

Dun, Fay A. (J) instructor in mechanical engineering department, Ohio State University, *Columbus, Ohio.*

Ellison, Luke V. (J) automotive service, Ellison Service, 222 North Main Street, *South Deerfield, Mass.*

Foltz, Harold H. (A) president and general manager, Foltz Truck Service Co., 78 North Summit Street, *Akron, Ohio.*

Kepler, Arthur R. (A) sales engineer, Stewart-Warner Speedometer Corporation, Chicago, (mail) 2309 Chester Avenue, *Cleveland.*

Stevens, S. B. (M) *Rome, N. Y.*

Sykes, George (A) general manager, Van Blerck Motor Co., *Monroe, Mich.*

Tilden, Merrill W. (A) president, Falls Motors Corporation, Sheboygan Falls, Wis., (mail) 163 West Washington Street, *Chicago.*

Tuttle, J. H. (M) chief engineer, Westcott Motor Car Co., *Springfield, Ohio,* (mail) 1003 Garfield Avenue.

Urban, William C. (A) chief engineer, Hoyt Metal Co., *Granite City, Ill.,* (mail) 2247 B Street.

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

INDEX TO VOLUME X, JANUARY-JUNE, 1922

Personal Notes of the Members

Items regarding changes in business connections, promotions, etc., are desired from the membership for insertion in these columns. This will enable members to keep their friends informed of their whereabouts and will also assist in keeping the records of the Society up to date.

Howard G. Benedict, formerly works manager of the Scott & Fetzer Co., Cleveland, has been made general manager of the A. J. Deer Co., Hornell, N. Y.

William E. Benninghoff has severed his connection with the Western Automatic Machine Screw Co., Elyria, Ohio, where he was time study engineer. He has not announced his plans for the future.

A. C. Bigelow has been elected president and general manager of the United Motor Car Co., Trenton, N. J. He was formerly industrial engineer for the Cook Linoleum Co., also of Trenton.

E. R. Birchard has been appointed factory representative for the General Motors of Canada, Ltd., Oshawa, Ont., with headquarters at Toronto.

Lloyd I. Birckelbaw has become associated with the Hudson Motor Car Co., Detroit.

Lloyd C. Boivin has been appointed shop foreman for the Central California Electric Co., Tulare, Cal.

Horace A. Brown, Jr., has been appointed manager of the sales division of motor bearings of the Hyatt Roller Bearing Co., Detroit, having been transferred from the motor bearings division, where he was also manager.

Porter A. Buck, previously chief engineer for the Parcol Industries, Inc., Alpena, Mich., has become associated with W. Buck & Sons, Berrien Springs, Mich.

Robert H. Campbell, vice-president and chief engineer of the Wharton Motors Co., Pittsburgh, is now located at the company's offices at Johnstown, Pa.

Oscar F. Carlson has formed the Oscar F. Carlson Co., with offices and factory at 562 West Washington Boulevard, Chicago. He was formerly chief engineer for the Amalgamated Machinery Corporation, also of Chicago.

R. E. Carlson, formerly chief of tank section of the tank, tractor and trailer division, Ordnance Department, City of Washington, has been appointed engineer in the Bureau of Standards, City of Washington.

At a recent meeting of the Cleveland Chamber of Commerce, **F. C. Chandler,** president of the Chandler Motor Car Co., who served last year on the board of directors, was reelected.

Walter P. Chrysler, who resigned as a director of the Republic Motor Truck Co., Inc., Alma, Mich., and a member of the executive committee at a recent meeting of the board of directors, will, in the future, devote all of his time that is available for automobile matters to the Maxwell Motor Co., Detroit, of which he is chairman of the board.

(Continued on page 4)

Personal Notes of the Members

Continued

DeWitt Clausen has severed his connection with the McVicker Engineering Co., Minneapolis, Minn. His future plans have not been announced.

Harold H. Coburn has resigned as director of industrial arts in the schools of Wellesley, Mass., where he has been installing machinery and building up the department for 5 years, to accept the position of purchasing agent and assistant to the president of the E. Van Noorden Sheet Metal Works, Roxbury, Mass.

O. F. Conklin has been engaged as consulting engineer of the Corcoran-Victor Co., Cincinnati. He was identified with the Remy Electric Co., Anderson, Ind., for a number of years, first as chief engineer and latterly as president and general manager.

H. Copleston, who for a number of years was assistant engineer and designer for the H. W. Johns-Manville Co., New York City, is now consulting engineer for Johns-Manville, Inc. Before returning to the Johns-Manville organization, Mr. Copleston was a sales engineer for the Doehler Die-Casting Co., Brooklyn, N. Y.

P. W. Cornelius is now proprietor of the Irvington Motor Service Co., Indianapolis, Ind. He was formerly salesman for the Buck Co., also of Indianapolis.

W. M. Corse, who on April 1 assumed charge of the division of research extension for the National Research Council, was elected a manager of the American Electrochemical Society at its recent annual meeting held in Baltimore.

Leroy V. Cram has severed his connection with the Mercer Motors Co., Trenton, N. J., where he was production engineer. His plans for the future has not been announced.

W. R. Davis has been promoted from sales and advertising manager to general manager of the Packard Engineering Co., Cleveland.

John C. Dayton, formerly engineer and manager for the Dayton Gas & Electrical Engineering Works, Harrisburg, Pa., has been engaged as traveling expert on engines, tractors and trucks for the International Harvester Co. of America, Chicago.

R. M. de Vignier has severed his connection as chief engineer for the American Vulcanized Fibre Co., Wilmington, Del. For a number of years he was development engineer for the Western Electric Co. and the du Pont Co. No announcement has been made regarding his plans for the future.

Edward H. Diesz has accepted the position of chief engineer with the United States Street Sweeping Machine Co., Akron, O. He was formerly assistant chief draftsman at the Akron, Ohio, plant of the International Harvester Co.

Clarence A. Earl was elected president and general manager of the Earl Motors, Inc., at the annual stockholders' meeting, held at Jackson, Mich.

J. O. Eaton, formerly president and general manager of the Torbensen Axle Co., Cleveland, and latterly president of the Standard Parts Co., also of Cleveland, has, together with a number of others, acquired the interest in the Torbensen company previously owned by the Republic Motor Truck Co., Inc., Alma, Mich. In the reorganization that followed Mr. Eaton was elected president.

Early last month **Frank L. Eidmann** sailed for Europe on a business trip that, it is expected, will last 6 months.

(*Continued on page 6*)

PERSONAL NOTES OF THE MEMBERS

Continued

A. W. Einstein, post-graduate student in highway transport course, University of Michigan, Ann Arbor, Mich., has accepted a position with the Service Motor Truck Co., Wabash, Ind., as a highway transport engineer and a special representative in the sales department.

R. C. Enos, formerly sales representative of the Torbensen Axle Co., Cleveland, was elected vice-president of the company in connection with the recent reorganization.

Gordon M. Evans has been appointed superintendent of planning at the Maxwell Motor Corporation, Detroit.

Murray Fahnestock has been appointed technical editor of the *Ford Car Trade Journal*, Milwaukee, in addition to being technical editor of the *Ford Owner and Dealer Magazine*, a position which he has held for many years.

Arthur N. Goodfellow has been elected treasurer of the Nice Ball Bearing Co., Philadelphia, with which he has been connected for some time.

Harold C. Gortzig has been made assistant to the superintendent of the Curtiss Aeroplane & Motor Corporation, Buffalo, N. Y. He was formerly assistant engineer for the Parenti Motors Corporation, also of Buffalo.

P. F. Hackethal, formerly chief engineer for the Paragon Motor Car Co., Cumberland, Md., has been engaged as consulting engineer for the Fox Motor Co., Philadelphia.

Herbert A. Hansen has accepted the position of assistant to the president with the Johnson Gear Co., Berkeley, Cal.

Herman Henkel has been made vice-president, treasurer and service director of the Garlock Sales Co., Lansing, Mich. He was previously proprietor of the Automotive Service Co., also of Lansing.

Elbert J. Jenkins, formerly sales-engineer for the Dowmetal division of the Dow Chemical Co., Midland, Mich., is now associated with R. E. Olds of Lansing, Mich., as president and general manager of the Oldsmar Mfg. Co., Oldsmar, Fla.

Lewis P. Kalb has accepted a position as engineer for the Continental Motors Corporation, Detroit.

P. J. Kent, until recently electrical engineer for the Willys Corporation, Elizabeth, N. J., has been engaged by the Zeder-Skelton-Breer Engineering Co., Newark, N. J., as electrical engineer on the development of the Zeder motor car.

E. H. Kessler, formerly associated with the Davis Sewing Machine Co., Dayton, Ohio, is now chief chemist of the Thresher Varnish Co., also of Dayton.

Joseph Kirschmann is owner and manager of Kirschmann & Co., Vienna, Austria.

E. B. Knowles is associated with the United States Asbestos Co., Lancaster, Pa., in connection with sales to the automotive industries. He was formerly secretary and general manager of the Staybestos Mfg. Co., Philadelphia.

J. M. Kroyer is no longer president of the Kroyer Motors Corporation, Stockton, Cal., but is located in Los Angeles.

Howard A. Miller has severed his connection with the Automotive Service Co., Rochester, N. Y., as junior partner. His plans for the future have not been announced.

William Moiselle has been engaged as manager of the wholesale department of the Packard Columbus Motor Co., Columbus, Ohio.

(*Continued on page 8*)

Personal Notes of the Members

Continued

J. H. Naiden has resigned as engineer and director of laboratories for the Prest-O-Lite Co., Indianapolis, and has entered into business with his brother to manufacture the Naiden battery, under the firm name of the Iowa Storage Battery Co., Des Moines, Iowa.

Through an unfortunate error it was announced in the April issue of THE JOURNAL that **W. N. Nakamura** was no longer a draftsman in the employ of the Dort Motor Car Co., Flint, Mich. This was due to insufficient data on a change of address blank received from Mr. Nakamura. He is at the present time still connected with the Dort organization in the capacity of mechanical engineer.

A. G. Partridge has been appointed to special sales work in the tire division of the B. F. Goodrich Rubber Co., Akron, Ohio. For several years Mr. Partridge was vice-president in charge of sales at the Firestone Tire & Rubber Co., and later was engaged in a similar capacity with the Star Rubber Co. His duties with the Goodrich organization will be largely confined to the development of sales.

H. E. Pengilly has become associated with the Triplex Machine Tool Corporation, New York City. He was previously mechanical engineer in the motor carriage section of the artillery division of the Ordnance Department, Washington.

Herbert C. Phelan, formerly direct factory representative for the Cornwell Quality Tools Co., Cuyahoga Falls, Ohio, has been appointed automotive instructor in the U. S. Veterans' Bureau Vocational School, Portland, Me.

W. H. Pratt is now associated with the Bearings Service Co., New York City. He was formerly sales engineer for the Klaxon Co., Newark, N. J.

M. L. Pulcher, vice-president and general manager of the Federal Motor Truck Co., Detroit, has been chosen to head the National Association of Motor Truck Industries that was recently formed.

M. H. Roberts has been appointed vice-president in charge of engineering of the Franklin Railway Supply Co., New York City, having previously been chief engineer of the company.

Harry M. Rugg, who for the past year has been director of educational extension at the Michigan State Auto School, Detroit, has just been made an assistant experimental engineer in the engineering department at the Dodge Bros. automobile factory, also located at Detroit.

Arthur A. Schupp, until recently engineer for the Wilcox Motor & Mfg. Co., Saginaw, Mich., has accepted a similar position with the Fredericksen Co., also of Saginaw.

M. Charles Schweinert, who for over 35 years has been associated with A. Schrader's Son, Inc., of Brooklyn, N. Y., as general manager, treasurer, director and president, has resigned. He has been retained by the corporation in an advisory capacity that will occupy a portion of his time and he contemplates devoting the remainder of it to other engineering and mechanical matters and to looking after his private interests.

L. S. Sheldrick has severed his connection as chief engineer and works manager for the Hicks-Parrett Tractor Co., Chicago Heights, Ill. His future plans have not been announced.

At the annual stockholders' meeting of the Earl Motors, Inc., Jackson, Mich., **William Sparks,** president of the Sparks-Withington Co., also of Jackson, was elected a member of the board of directors.

(*Concluded on page* 10)


Midwest engines are manufactured under patents covering a lubricating system, which is the only system the industry knows today that will secure these results:
-Exceptional torque and horse power at both low and high speeds
-elimination of oil pumping which combined with proper cooling permits higher compression
-greater performance ability at the business end of the engine in the elimination of internal friction
-long lived economical performance with absolute minimum of maintenance, attention and expense.
These are the reasons why Midwest engines are today used in many leading trucks, tractors, motor buses, railway motor cars and industrial locomotives.
When will your product be in the picture?
Lon R. Smith
LON R. SMITH, Vice-President
MIDWEST ENGINE COMPANY - - INDIANAPOLIS, U. S. A.
MIDWEST
TRUCK and TRACTOR ENGINE

PERSONAL NOTES OF THE MEMBERS

Concluded

Glenn A. Toaz, formerly assistant chief engineer for the Butler Mfg. Co., Cleveland, has been made service engineer for the Steel Products Co., also of Cleveland.

G. W. Turney has severed his connection with the Rome-Turney Radiator Co., Rome, N. Y., where he was treasurer and general manager. He has not announced his plans for the future.

J. D. Van Vliet is no longer a chief engineer for the Cantilever Aero Co., New York City. His plans for the future have not been announced.

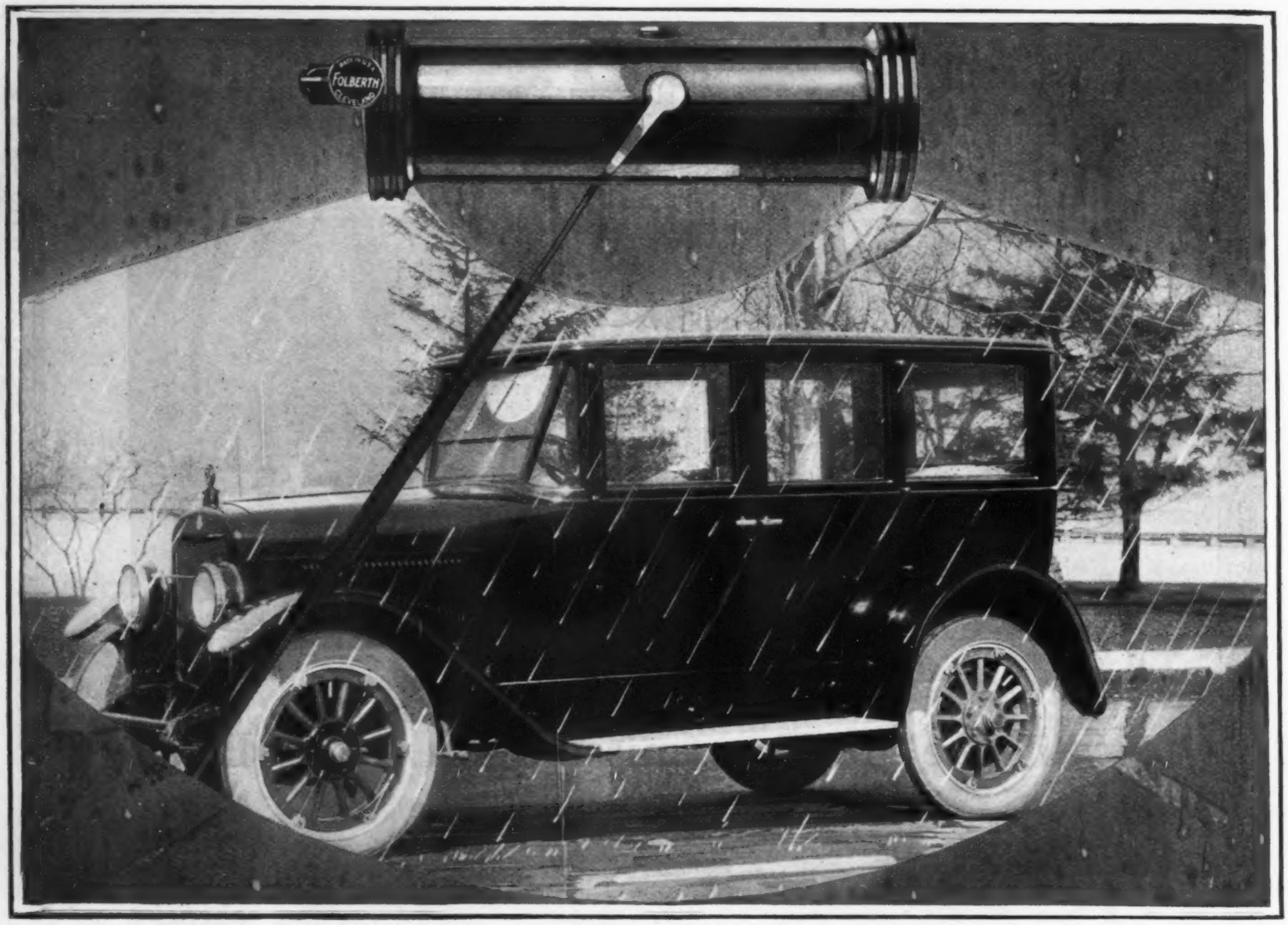
R. O. Watson has severed his connection as service manager for the White-Humphreys Motor Co., San Francisco, and is now associated with S. Markowitz, also of San Francisco.

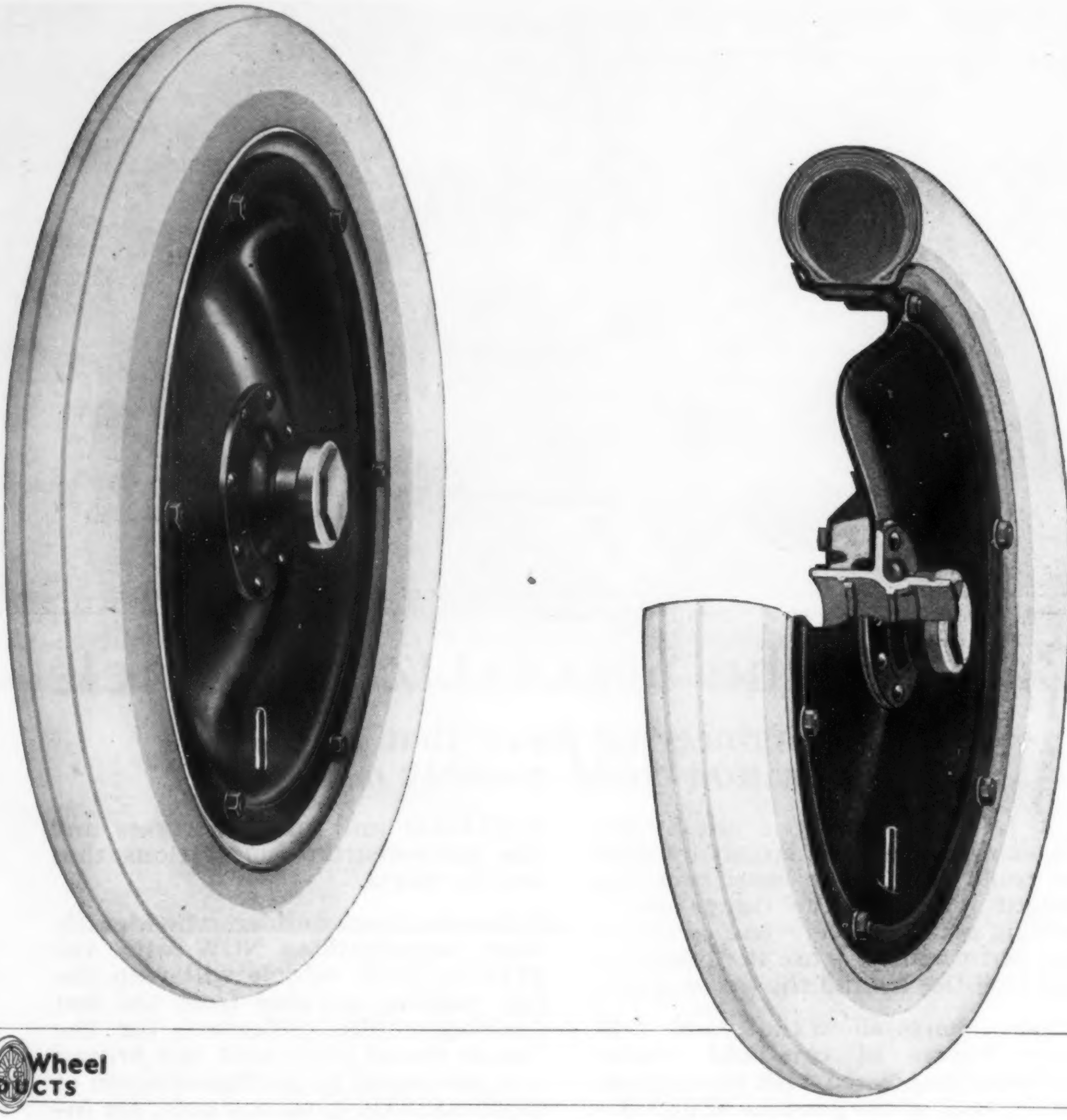
Paul St. Elmo Webb has been appointed sales promotion manager of the Hinkley Motors, Inc., Ecorse, Mich. He was formerly a transportation engineer with the Diamond T Truck Co., Nashville, Tenn.

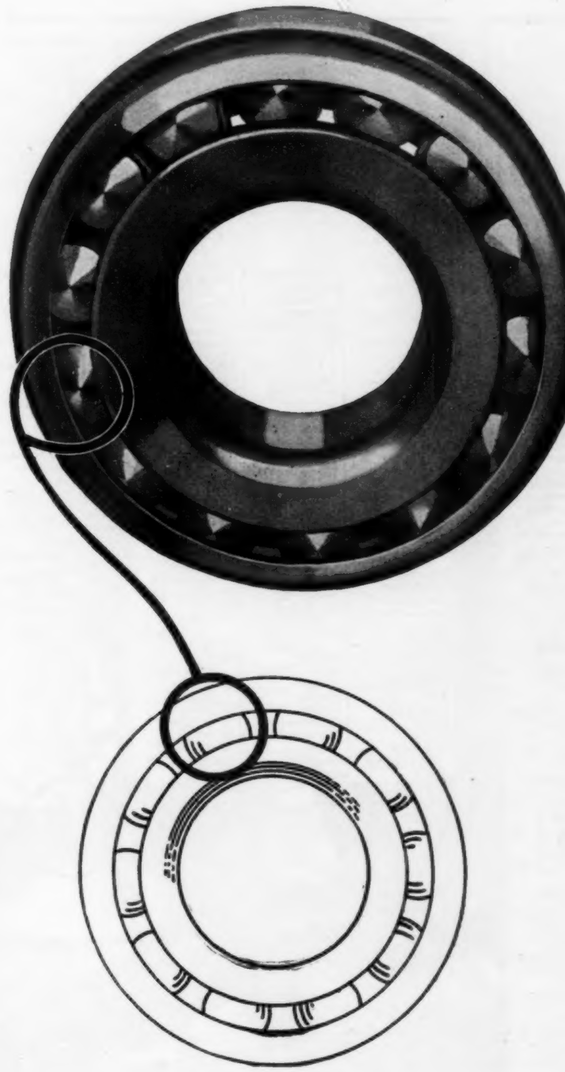
George William Winter, formerly foreman at the Avondale Motor Car Co., Cincinnati, has opened an automotive repair electrical service under the name Winter & French, also at Cincinnati.

Walter H. Woods has been transferred from the motor bearings division of the Hyatt Roller Bearing Co., New York City, where he was district manager, to the Detroit office of that company in the General Motors Building.

K. W. Zimmerschied, president of the Chevrolet Motor Co., Detroit, has been appointed assistant to the president of the General Motors Corporation, retaining however, the presidency of the Chevrolet organization.


FOLBERTH





Frictionless As Balls By Actual Test

An experiment has been recently completed by the manufacturers of a well known electrical vehicle in which Bock Taper Roller Bearings were placed in competition with one of the best makes of ball bearings.

The Bock rollers proved equally efficient as the ball, the consumption of current required to operate on either type of bearing being virtually the same, in some particulars actually showing a slight advantage for Bock.

Such remarkable efficiency is accounted for by the *spherical headed roller design*, an exclusive Bock feature, and the extraordinarily precise manufacturing processes employed.

This experiment proved conclusively that the Bock Patented Taper Roller Bearing *combines* the efficiency of the ball bearing with the ruggedness, increased capacity and adjustability of the taper roller bearing.

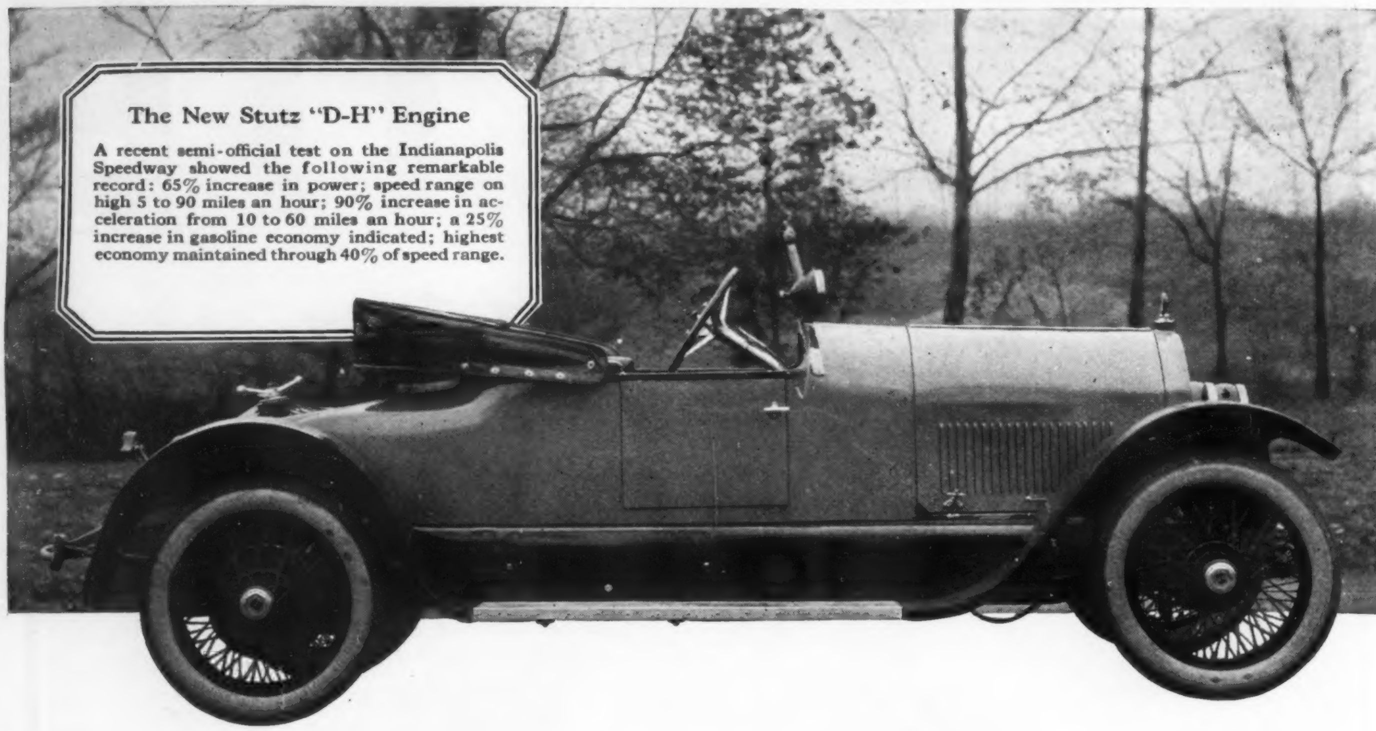
THE BOCK BEARING COMPANY
TOLEDO, OHIO

BOCK

Quality TAPER ROLLER BEARINGS

MEMBERS' PROFESSIONAL CARDS


A good bearing
It is not difficult to make a ball bearing. But a *good bearing* is a distinct achievement. The Schatz Universal Annular Ball Bearing embodies exclusive features of design and the highest manufacturing practice.
Specify the bearing with a margin of safety—Schatz Universal.
The Federal Bearings Co., Inc.
Poughkeepsie, N. Y.
Schatz
UNIVERSAL
Annular
BALL BEARING



New Stutz Motor, Mogul-Equipped Shows 65% Increase In Power

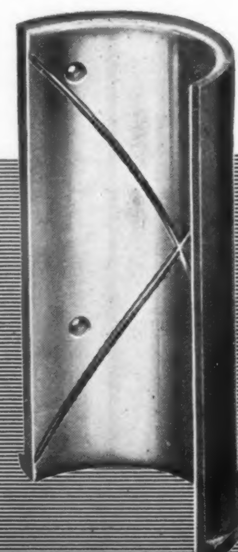
The Stutz is another of these outstanding, rugged American cars that have firmly entrenched themselves as standard values in the world of fine automobiles.

The sensational performance and success of the new Stutz models with the new "D-H" engine, can only be due to perfect engineering, material, parts and workmanship.

Mogul Motor Bearings are standard equipment on these powerful engines as they have been on previous Stutz engines. In fact Mogul Motor Bearings are invariably found associated with power plants of long-standing, recognized merit. And what is more significant is the fact that Mogul Bearings have gained such recognition not only in the automotive field, but likewise in the aircraft, marine and stationary engine fields.

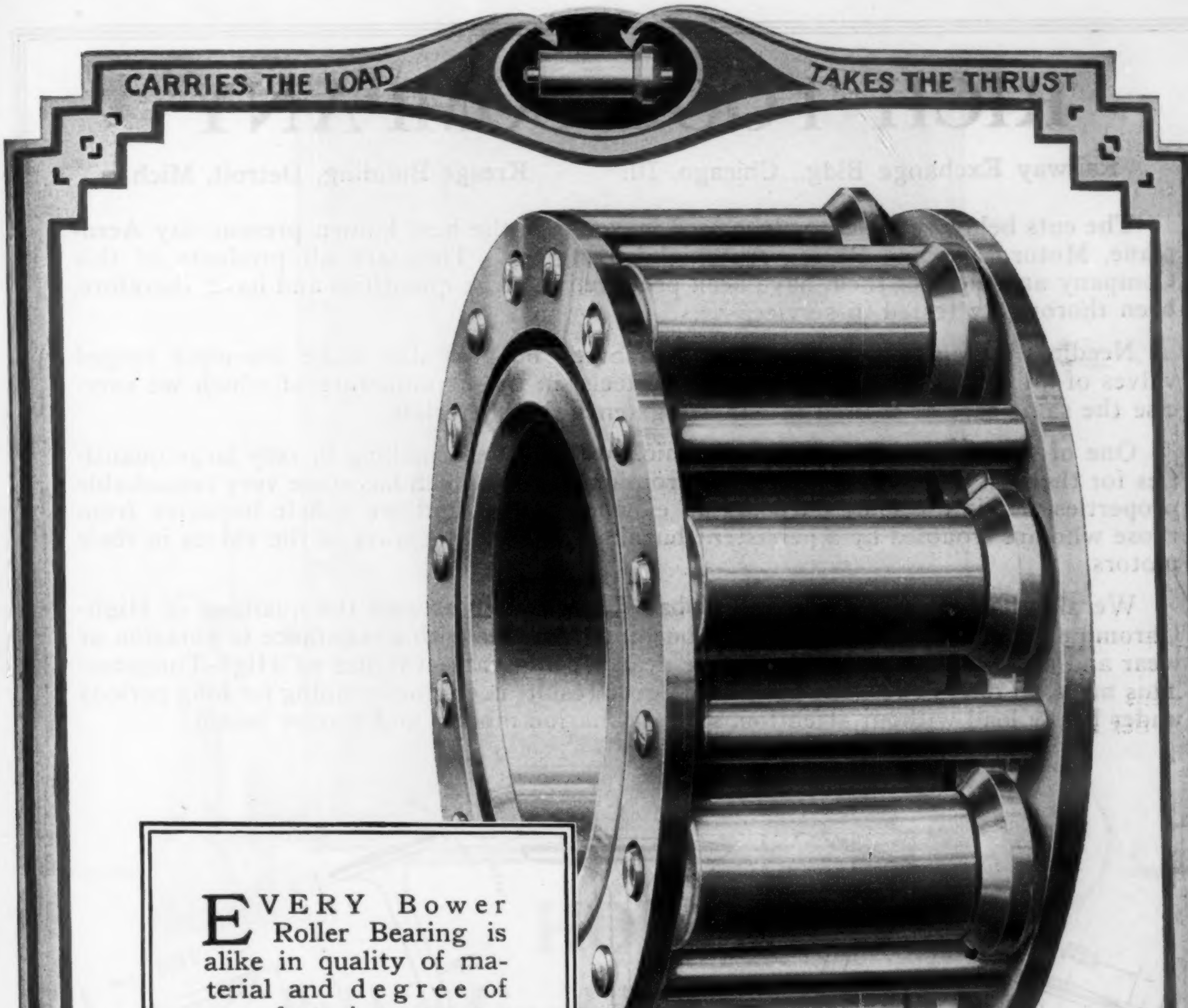
It stands to reason that motor bearings that have made good in every field in which motor bearings are used, must have genuine merit. Automotive engineers choose Mogul Motor Bearings purely on their well known records of performance.

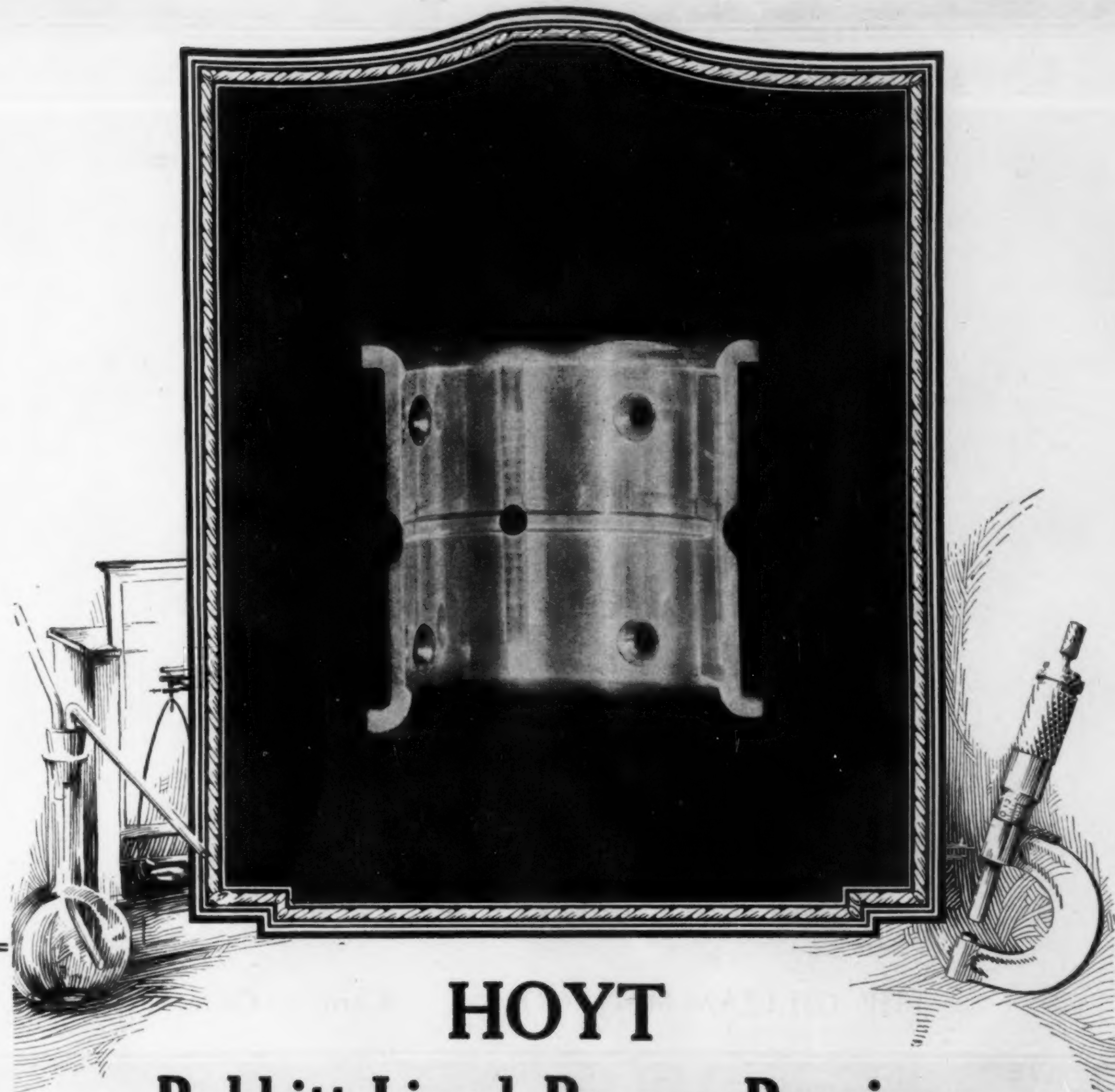
MUZZY-LYON COMPANY
Detroit, Michigan



Mogul

Bearing Alloys *and* Finished Bearings

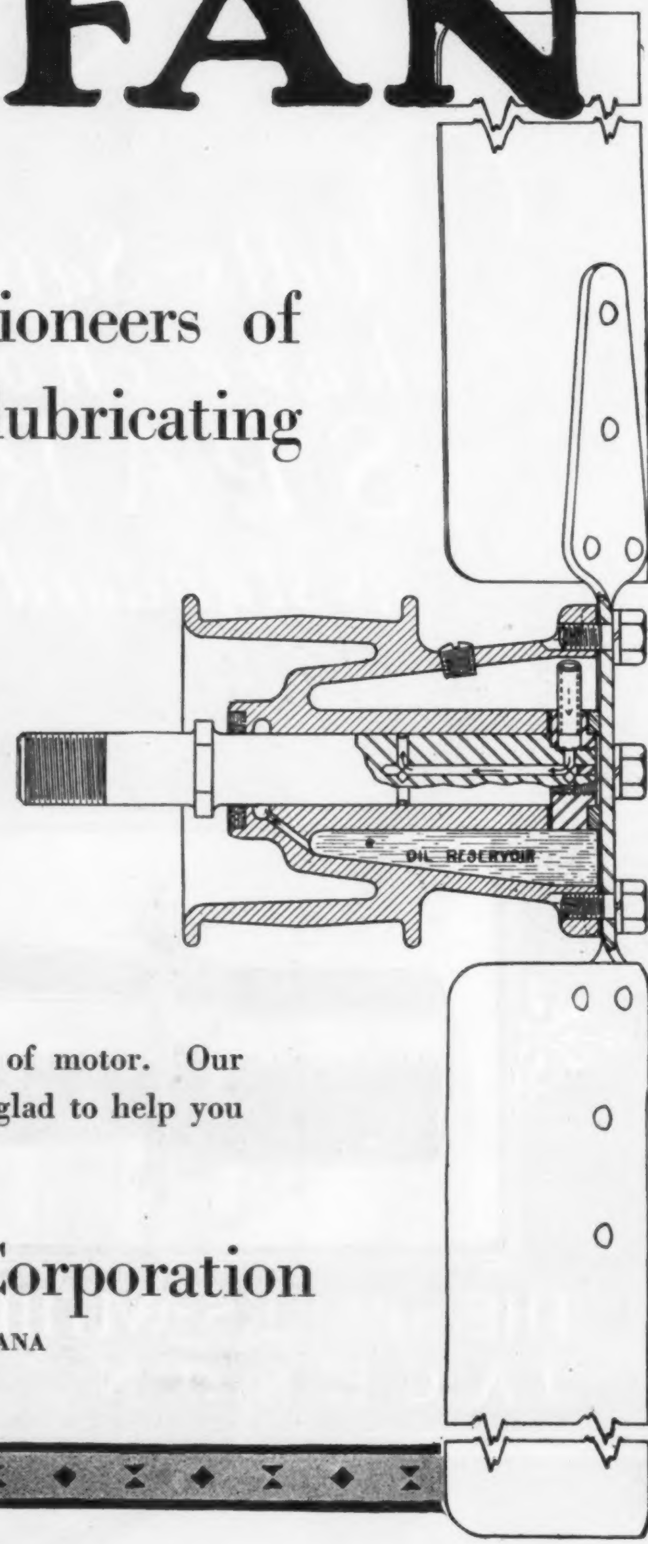

CARRIES THE LOAD
TAKES THE THRUST
EVERY Bower Roller Bearing is alike in quality of material and degree of manufacturing accuracy.
Pre-determined standards and careful inspection regulates that.
BOWER
ROLLER BEARING CO.
Detroit Michigan
Exclusive Bower Features
Separate bearing surfaces for load and thrust. Parallel raceways. Self-aligning. Never need adjusting. Does not develop end thrust under loads. Will not bind or end-slip.
BOWER SAVES POWER




Wills-Sainte Claire have chosen the SPARTON as standard equipment
THE SPARKS-WITHINGTON COMPANY
JACKSON, MICHIGAN

SERVICE FAN

FLOOD OILING



Originators and pioneers of plain bearing, self lubricating fans

Service fans retain their oil by mechanical means and do not rely on washers. Every moving part is lubricated under pressure, thus insuring practically no wear. The result is a silent bearing, the fan giving maximum cooling with a minimum of power consumption.

Patent Oct. 20, 1920.

We have a fan for every make of motor. Our engineering department will be glad to help you with your cooling problem.

Service Products Corporation

Indianapolis, Indiana

The Six Factors in Raulang Responsibility

TRADE MARK

The Third Factor— Our Low Costs

OUR ability to pay cash, plus our time-built knowledge of what to buy (we have been coach builders for more than sixty years) materially aids in combining remarkably high value with remarkably low costs.

For well over 60 years we were builders of fine coaches and later of electric automobiles—a rich background of experience that has stood us in good stead in building fine bodies on a quantity production basis. The fund of knowledge thus attained is necessarily beyond the reach of the manufacturer launching right into body building from the first.

This 60 years of Know-How is at once your protection and your profit—(and it is incidentally the reason for our fine coach work—the finest in the body field).

RAULANG BODY DIVISION
The Baker R & L Co.
CLEVELAND, OHIO. U.S.A.

Raulang
TRADE MARK
AUTOMOBILE
BODIES


Spicer
Propeller Shaft
Sheldon
Springs for Motor Cars and Trucks
Sheldon
Axles for Motor Trucks
SALISBURY
Axles for Passenger Cars
Parish
Heat Treated Frame

S. A. E. Employment Service

The following announcements are published for the benefit of members of the Society and the convenience of companies in need of men. No charge whatever is made for this service. In the case of items prefixed by an asterisk further information is withheld at the request of the company or individual making the insertion, but written communications bearing the number of such items will be forwarded by the office of the Society. Applications for positions from non-members must be endorsed by a member of the Society.

No announcement will be repeated in these columns unless specific information is at hand that the respective member is available or the position unfilled.

To put available men and employers in touch with each other quickly, this column is supplemented twice weekly by a bulletin giving the latest information with regard to Men and Positions Available. The bulletin will be mailed to members and employers on request.

It is suggested that for quick action in securing men or positions there be specified for publication the name and address, a post-office box number or a general mail delivery address. If applications are confidential, the Society will forward replies through an index number.

For the good of the service members securing or filling positions or companies securing men through these columns are urged to send advices to this effect to the office of the Society promptly.

Considerable time can be saved in securing a position if the following items of information are given in registering with S. A. E. Employment Service.

1. Date
2. Name in full
3. Present mail address
4. Telephone number
5. Present telegraph address
6. Permanent mail address
7. Telephone number
8. Married or single
9. Nationality
10. Age
11. Height
12. Weight
13. State of health
14. Education (Give schools and colleges attended, courses taken and duration of each course)
15. Previous employers (Give names, addresses, term of employment with each, your title and the nature of the work done)
16. Are you employed at present?
17. References (Give names and address of three or four persons not related to you and with whom you have been directly associated)
18. What societies, clubs or associations are you a member of?
19. Salary wanted per week, month, year
20. What general classifications do you wish to be listed under? (List in order of preference. For example: General Manager, Works Manager, Research Engineer, Production Engineer, etc.)
21. Do you wish your name and address published in your announcement?
22. Do you wish to use a post-office box number? (If so, arrange with your postmaster for the use of a box and send the number of the box to us, giving town and state)
23. Is your name to be withheld from publication? (In this case your announcement will be run under an index number and replies will be forwarded through the office of the Society)
24. Prepare copy for insertion in THE JOURNAL and Bulletin. (Make it as complete and as concise as possible)
25. Any preference as to location?
26. When will you be available?

MEN AVAILABLE

0364 EXECUTIVE ENGINEER with 20 years' experience on automobiles, including both engines and finished cars. Has held position of works manager. Able to layout large plant, organize and handle working force, cut costs, get maximum production and highest standard of accuracy. Available at once. No preference as to location.

0365 SALES ENGINEER AND EXECUTIVE with 15 years' experience in the automotive accessory field, wishes to connect with an accessory manufacturer where his experience would be of value. Has held responsible positions.

0366 AUTOMOTIVE ENGINEER College graduate with 6 years' experience as designer, checker, chief draftsman and engineer principally on engines. Especially familiar with marine engine and automobile design and thoroughly experienced in production methods. Available immediately. Age 27.

(*Continued on page 60*)

S. A. E. EMPLOYMENT SERVICE

Continued

MEN AVAILABLE

0371 ENGINEER with design for a radial air-cooled engine with patented features, wishes to communicate with persons interested in its development.

0372 ENGINEER with 23 years' experience in various lines including machine tools, steam and internal-combustion engines, tool and jig making and designing, automobiles and aircraft engines. Has been connected with a number of large automobile builders in executive capacity. At present employed by the Government in charge of engine maintenance throughout the Air Service. Location, immaterial. Age, 38; married.

0373 ASSISTANT ENGINEER or designing draftsman, with varied experience in charge of truck engineering drafting, tool design and automotive machine work. Formerly chief draftsman with company building high-grade motor trucks. Technical graduate. Age, 27; married. Available at once.

0374 SALES ENGINEER with headquarters at Salt Lake City desires to represent Eastern manufacturer or wholesaler in the intermountain territory.

0376 CHIEF ENGINEER, research engineer or factory manager. Seven years' practical experience in mechanical and electrical engineering in automobile and allied industries. At present assistant to chief engineer of prominent motor-car company. Experienced in research, service, experimental and development work; also in consulting work pertaining to construction and design of automobile, marine or aviation engines.

0377 ENGINEER AND PRODUCTION EXECUTIVE with over 12 years' experience on high-grade automotive engines and transmissions. Has recently tooled and put into production one of the highest grade eight-cylinder engines on the market. Experience covers designing, tooling, production, purchase of materials and costs. Able to build up organization and produce quality material at minimum costs. Best of references. Available on 30 days' notice.

0389 ENGINEER An original, thorough and resourceful practical man, with 16 years' experience in experimental work, development, testing, service, inspection and factory problems. Age, 35; single.

0390 MECHANICAL DRAFTSMAN with 8 years' experience in design and computations and 4 years' university work in mechanical engineering and chemistry desires a position. Prefers designing or testing of internal-combustion engines and a location that is favorable to the continuation of evening study at a university or college. Age, 30; single. Best of references can be supplied.

0395 MECHANICAL ENGINEER with 14 years' experience as a designer, machine-shop foreman, chief draftsman and efficiency engineer, also broad experience in the designing of special machinery, tools, jigs, fixtures, gages and dies, desires a position where initiative and inventive ability along manufacturing lines are considered. Married; age, 34.

0396 TECHNICALLY TRAINED YOUNG MAN with some business experience desires to make a permanent connection with an automobile company in the sales promotion department. Single. Available immediately. No preference as to location.

0397 ENGINEER with 5 years' experience in the automotive industry desires a position with an automobile, truck, tractor or engine builder or a large garage company. Possesses executive ability and has held positions as manager and purchasing agent. Age, 26; single. Available at once.

0398 WORKS MANAGER A practical man with initiative and ability who can get results. Broad experience in factory and engineering management including automobile, aeronautic and steam engine units of various kinds. Age, 32; unmarried. Available immediately. Eastern location preferred.

(*Continued on page 62*)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE
Continued

MEN AVAILABLE

0399 DRAFTSMAN Energetic hustler desires to make a connection in New York or New England States, but would consider other location. Is thoroughly familiar with passenger-car chassis work and punch and die work for automobile stampings and is able to work with engineer. Married; age, 25. Available immediately.

0403 ENGINEER who has been 27 years in automotive industry. A technical graduate with experience in development work, as chief engineer and as an executive. Is a progressive organizer and a thorough exponent of cooperation. Could furnish competent department force if required. Has developed and conclusively proved many items of betterment, is fully awake to present market conditions and is prepared to do his part in meeting them. Age, 47. Address H. C. Gibson, 2304 Seventh Avenue, Moline, Ill.

0408 MAN with 15 years of real mechanical engineering in the automotive field would consider special work such as a consulting, designing and chief engineer in the marine and industrial engine fields. Is familiar with the problems of building mechanical confidence and new business. Married. Available now. Location immaterial.

0411 ENGINEER who has devoted the past 12 years to the automotive field principally in experimental and production work is available for a position. Is technically educated and has held the positions of chief inspector, general foreman, assistant production manager, production manager and superintendent with companies producing motorcycles, passenger cars, trucks and tractors. Has also designed tools, jigs, fixtures and gages. Location immaterial. Married; age, 32.

0412 AUTOMOTIVE ENGINEER with 7 years' experience in truck and tractor design desires a position with some established truck and tractor organization. Age, 25; married. Middle West preferred but not essential. Available at once.

0413 DESIGNER AND DRAFTSMAN desires to establish a connection with a manufacturer along the lines of production and technical development. Is a graduate engineer with experience in marine engineering, both theoretical and in actual service; Diesel engines a specialty. Age, 25; unmarried. Available immediately. Vicinity of New York City preferred but not essential.

0414 YOUNG MAN wishes to become associated with a large progressive builder of open or closed passenger-car bodies. Is willing to start in a middle-rate position as a draftsman with an opportunity to be in contact with high-grade aggressive men and where, when worth is proved, advancement would follow to a higher and more responsible position. Two years' college education in mechanical engineering and 1 year's experience in body drafting. Age, 24.

0416 CHIEF ENGINEER of a prominent automobile company seeks to establish a new connection. Was technically educated in Europe and has had 12 years' experience in automotive design and experimental work; also 4 years as chief engineer for a large engine builder. Would accept the position of chief, assistant chief or experimental engineer; undertake special development work or also consider a position as technical representative in Europe. Original and uptodate. Possesses a knowledge of French and German. Age, 34; married.

0424 ENGINEER AND EXECUTIVE with 8 years' experience in the automotive industry. Has been associated with leading companies and industrial and consulting engineers. Is especially experienced in handling men and installing systems as sales engineer and is qualified to assume a position of responsibility. Age, 27; married. Available now. New York City preferred but not essential.

0426 SERVICE ENGINEER with over 16 years' experience in all departments as executive, organizer and efficiency expert in passenger-car and truck details. Is also experienced in promotion work, the standardization of service-station equipment and methods, the installation of flat price cost system on repairs, etc., and has had truck factory organization experience.

(*Continued on page* 64)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE
Continued

MEN AVAILABLE

0427 MECHANICAL AND ELECTRICAL ENGINEER Technical graduate with experience in motor-truck layouts, jig and fixture design is now available for a position. Experienced also in checking, preparing specifications, shop-ordering, molding and drafting. Age, 28; married. Atlantic coast preferred but will consider any location.

0428 MECHANICAL ENGINEER who is also an expert draftsman and designer desires to make a Detroit connection in the capacity of selling, cost or production executive. Is experienced in purchasing and management. Age, 28; single. Available on 2 or 3 weeks' notice.

0429 GENERAL MANAGER wants an opportunity where ability, experience and energy are needed to form a new organization, or to carry forward or develop further one already formed. Has developed present plant from its inception to a modern, well-equipped, well organized and efficiently operating organization and has been chief engineer with a large organization for 10 years. Experienced in all branches of automotive work as well as gas engine construction. Available immediately if necessary.

0430 AUTOMOTIVE ENGINEER College graduate with 3 years' experience as designer, checker and chief draftsman and familiar with automobile engine, axle and chassis design. Age, 23; single. Available at once. Location, immaterial.

0432 ENGINEER with college training desires a position. Experienced in automotive design, sales engineering, drafting, chassis design, inspection, maintenance and repair. Available immediately. Age, 28; unmarried. Location, immaterial.

0433 PRODUCTION OR TOOL EXECUTIVE with 13 years' practical experience in production and tool work on automobile accessory wishes a position with a reliable company where merit is recognized by promotion. Age, 34; married. Available on 15 days' notice. Central states preferred but not essential.

0437 GRADUATE MECHANICAL ENGINEER, with 5 years' testing, development, production and operation experience; also business experience. Knows languages and possesses a sense of adaptability for new duties. A hard and serious worker. Age, 29; married. Available now. Location, immaterial.

0442 CHIEF ENGINEER OR MANAGER Nine years' executive factory experience in organization and management. Has made a specialty of tractors, both the wheel and track-laying types, transmissions, internal-combustion engines, railroad motor cars and gasoline and kerosene locomotives. Age, 39; married. Location, immaterial.

0443 RESEARCH ENGINEER Graduate engineer with 8 years' mechanical and electrical technical experience, followed by post-graduate study at a leading university, desires a position. Is capable of conducting pioneer investigations and development work and has had 2 years' automotive design experience. Age, 30. Vicinity of New York City preferred.

0444 DESIGNING ENGINEER is open for engagement with a progressive company. Is experienced in the design of internal-combustion engines, on experimental truck, tools, tanks, tractors and trailers; also in drafting, maintenance and repair. Has held an executive position. Age, 33; married. Available on short notice. Location immaterial.

0445 MECHANICAL ENGINEER Experienced in transmission layout, drafting, checking drawings and patterns, and chassis design. Has been in charge of tool designing and inspection; also held the positions of assistant foreman in machine shop and mechanical engineer. Technical education. Age 34.

0447 BALL BEARING ENGINEER having sales promotion and designing experience is available for a position. Is an engineer with 9 years' experience, including 5 years' specialization on the design and promotion of ball bearings and 3 years in an executive capacity. A man who is particularly interested in a position affording opportunities toward sales and advertising management and where past experience and knowledge of the business combined with character and personality count. Age, 27; married. Available within reasonable time.

(*Continued on page* 68)

See announcement at the head of the S. A. E. Employment Service column, page 58

S. A. E. EMPLOYMENT SERVICE

Continued

MEN AVAILABLE

0451 Engineer with wide experience, fully qualified in engineering, production, or factory management available for position. Age, 38; married. Available at once. Location immaterial.

0453 Graduate Mechanical Engineer with 4 years' automotive engineering experience and inventive ability desires a position in research laboratory or with company developing some new engine or car. Would consider teaching at some institution having a well-equipped research laboratory. Was formerly instructor in automechanics with Government; also instructor in automotive engineering and mechanical engineering. Has had gasoline engine design and testing experience. New York City preferred but not essential. Age, 27; single. Available on short notice.

0457 Sales Executive with long experience in connection with automotive ignition, lighting and starting apparatus. Familiar with production methods. Technical graduate and well known in the automotive industry. Age, 31; married. Available within a reasonable time.

0460 Metallurgical Engineer Technical graduate with 10 years' practical experience in physical and chemical testing, inspection, metallurgical research and heat-treatment of all grades of steel. Competent to install heat-treating plant and laboratories. Capable executive. Age, 37; single.

0461 General Automotive Construction Engineer, with practical experience, desires a position. Age, 31; married. Ex-service man who has been connected with large truck and passenger-car plants.

0463 Technical Graduate in electrical engineering desires position in or near New York City with electrical or automotive company that offers an opportunity for advancement. Ready and willing worker; aged, 25 years. Last position was of executive nature with machinery export house. Available July 1.

0464 Branch Manager and Engineer, associated for past 14 years with an internationally prominent ignition company. Has had 20 years' practical experience with internal-combustion engines of all kinds, including 10 years' active sales and engineering experience as branch manager. Would like to enter some new field where combined engineering and selling experience would be applicable. Good references furnished.

0465 Service Man with 17 years' experience in service and experimental work on motor cars, wishes to make a connection with first-class organization, on the road or in a service station. Age, 36; married. Available immediately. New York City or vicinity preferred.

0466 Man with factory and garage experience, including 2½ years' handling fleets of trucks and nearly 4 years in the Government's largest training schools as automotive instructor, is available for a position. Prefers to locate with firm operating large fleets of trucks or with factory as a service man. Age, 28; single. Can furnish the best of references and qualifications. Location immaterial. Available in 20 days.

0467 Engineer Technically trained man with 10 years' experience as factory and engineering executive both in automotive as well as in electric machine-tool plants, desires similar position with progressive, uptodate manufacturing organization, where ability to produce is appreciated. Age, 29; married. New England or Eastern States preferred, but will consider a good proposition elsewhere. Address O. H. L., 217 Midland Street, Bridgeport, Conn.

0468 Mechanical Engineer wishes position with a reliable company. Has had 4 years' experience in the automotive industry as layout draftsman; also some shopwork. Age, 27. Excellent references. Available June 15.

(Continued on page 70)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE
Continued

MEN AVAILABLE

0471 SALES OR SERVICE ENGINEER on magneto or battery systems. Seven years' practical road work.

0472 STOCK SUPERVISOR Experienced engineer with several years' experience in handling, storing and issuing stock to the factory and the follow-up of materials, both in the factory and from outside vendors. A permanent position with a live going company is desired. Available at once. Location immaterial.

0473 SALES EXECUTIVE with over 15 years' experience in the automotive field, wants a parts or accessory proposition in the Middle West having offices in Chicago and Cleveland. Technically educated and well known in the trade. Past record is one of successful sales and development of territory. Only a permanent connection with a big future considered. Age, 33; married.

0476 YOUNG ENGINEER Civil and electrical engineering graduate wishes to reenter the automotive field. Has had over 2 years' experience on all phases of steam automotive development. Hard worker. Location immaterial. Employed at present but available on short notice.

0479 SALES ENGINEER AND SERVICE EXECUTIVE A college trained man of broad practical experience and successful record would like to get in touch with progressive automotive manufacturer who has an opening for a man of initiative and ability. Age, 28; married. Middle West preferred but not essential.

0480 AUTOMOTIVE ENGINEER Technical graduate with 15 years' experience in truck engineering desires a position as engineer or sales engineer with truck or truck equipment company. Available immediately. Married; age, 41. Middle West preferred.

0483 CHIEF DRAFTSMAN Technical graduate. Age 26; experience covers the design and development of tools, machinery and equipment for economical production; also factory management, including maintenance and general construction and 2 years' selling and investigating on machinery and gray-iron castings. Age, 25; unmarried. New York City or vicinity preferred but not essential.

0484 MECHANICAL ENGINEER Technical graduate with 7 years' experience covering the design of motor cars, motorcycles, aircraft, pressed steel products, rubber machinery and ordnance. For 3 years was in charge of engineering department, supervising design, estimates and maintenance. Age, 29; married. Location immaterial. Available at once.

0485 MECHANICAL ENGINEER Factory executive with experience in production methods and quality at minimum cost. Can handle men and get cooperation and results. Has held executive positions for a number of years. Available at short notice. Age, 32; single.

0486 PRODUCTION AND MANUFACTURING EXECUTIVE Graduate mechanical engineer with thorough practical experience, including the completion of a 4-year machinist and tool maker apprenticeship, executive experience as superintendent, production engineer; consulting engineer on shop methods and production and organization problems for large industries desires connection with automobile, truck or parts organization where ability, experience and energy are required. Detroit preferred. Available on short notice.

0489 CHIEF INSPECTOR OR SERVICE MANAGER is available for a position. Twenty-two years' experience along this line. Cleveland or Buffalo preferred but not essential.

0491 LAYOUT MAN AND DESIGNER with 5 years' automotive design experience on motor-truck chasses and ball-bearing installations in passenger cars and trucks desires to connect with automobile or truck company. Vicinity of New York City, Newark or Jersey City preferred.

(*Continued on page* 72)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE
Continued

MEN AVAILABLE

0492 MECHANICAL ENGINEER Fifteen years' experience designing and constructing passenger and commercial vehicles. Has made a specialty of engine design. Prefer New York City or vicinity.

0499 MECHANICAL ENGINEER University graduate, with 10 years' automotive experience possessing considerable inventive ability and designer of several successful engines including the air-cooled type, desires position with company interested in developing a light-weight air-cooled car. Is competent to take charge of engineering, design and experimental work. Age, 34; married. Location, immaterial.

0500 SHOP SUPERINTENDENT OR CHIEF INSPECTOR in a truck assembly plant. Twenty-one years' experience.

0501 MECHANICAL AND AUTOMOTIVE ENGINEER with 3 years' experience in the shop and drafting room, a keen student in industrial and shop management problems and possessing a thorough knowledge of the various production methods. Is capable of acting in the capacity of assistant superintendent. Age, 22; single. Detroit or vicinity preferred but not essential.

0502 ENGINEER Eighteen years' experience in the design and production of automobiles including passenger cars and trucks. Age, 33; married.

0503 ENGINEER with 14 years' automobile engineering, designing and supervision experience. For past 2 years has held position of assistant engineer of prominent automobile company. Available immediately.

0504 ENGINEER Ex-service man and college graduate experienced in experimental work, design and manufacturing. Specialty, the elimination of both manufacturing and service difficulties, the reduction of manufacturing costs and the use of salvage material that would ordinarily be scrap. Would also consider positions of efficiency engineer, assistant engineer, sales engineer for gas engine, automobile or accessory manufacturer. Married; 28 years old. Best of references guaranteeing ability, character, integrity, energy and initiative can be furnished.

0505 MAN with highest grade engineering and business education, combined with twelve years' experience in engineering and management desires position. For past 6 years has held position of manager of successful manufacturing company and is experienced also in design of mine structures, surveying, investigation, and construction. Record will bear closest inspection. Age, 37; married. Available immediately. Location immaterial.

0506 AUTOMOTIVE ENGINEER Mechanical engineering graduate with 5 years' experience in testing, development, tooling and production organization, operation, maintenance and repair; also drafting and design and some business training. Has held executive position. Good references. Age, 29; married. Available at once. Location immaterial.

0516 ENGINEER wishes to accept the responsibility for the design of some part of a car such as axles and spring suspension, or to act as sales engineer to adapt the units of a parts maker to requirements of possible and actual customers. Experienced along both lines including both passenger cars and trucks, and specializing on axles, engines and electrical equipment. Has an extensive acquaintance among engineers and is accustomed to technical writing and correspondence. Available for moderate salary. Location immaterial. Address: Engineer, P. O. Box 317, Hartford, Conn.

0517 ENGINEERING EXECUTIVE An S. A. E. and A. S. M. E. member is available for position. Energetic and resourceful with over 20 years' experience in design, development work, tooling and production of automotive and other mechanisms. A designer of a successful, high quality

(*Continued on page* 74)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE

Continued

MEN AVAILABLE

passenger car and skilled in interchangeable production. Has held positions of chief tool and machine designer, experimental development engineer and chief engineer. Wishes to connect with well-established organization where creative and organizing ability, initiative and good address are required. Knowledge of French and German. Married. Available now.

0518 METALLURGICAL ENGINEER possessing strong and specific qualifications and competent to fill responsible position as engineer and executive desires to communicate with company requiring the services of such a man. Has held positions of metallurgical engineer and sales engineer and has been factory manager for past 4 years. College graduate. References of the highest order. Full details as to experience and qualifications to companies interested.

0519 LABORATORY ASSISTANT with 6 years' experience at general laboratory work, engine development and road testing. Best of references. Married. Location New York City or Cleveland but not essential.

0520 YOUNG ENGINEER Formerly chief draftsman of truck company with 10 years' experience in engineering department. Specialty, design, layout and engineering specifications. Married; age, 30. Middle West preferred.

0525 CHIEF DRAFTSMAN with 10 years' drafting room and several years' shop experience on passenger cars, trucks, and tractors is available for similar position or assistant engineer with automobile or accessory company. Wide experience on chassis design including engine, transmission and axles. Well versed in theory as well as practice. Middle West location preferred. Age, 31; married.

0527 DESIGNER AND DRAFTSMAN Technically educated man with 3 years' experience in truck, tractor, gasoline engine and ball bearing design. Prefers permanent position as junior sales engineer or position leading up to same. Chicago or vicinity preferred. Age, 25.

0531 AUTOMOTIVE ENGINEER with 17 years' experience in designing and building passenger cars, trucks, tractors and engines of heavy-duty and aircraft types. Was recently engaged in the design and construction of large heavy-duty air-cooled engines for tractors. Possesses a great quantity of valuable data on air-cooling. Technical graduate. Best of references. Salary a secondary consideration. Age, 32; single. Eastern location preferred.

0533 AERONAUTICAL ENGINEER with 9 years' experience in the design, construction, and production of aircraft. Thoroughly familiar with shop practice. Energetic; possesses executive ability and originality. Location, West preferred but not essential. At present with prominent airplane company in the East, but wishes to make a change. Technical graduate. Age, 39; married.

0534 TOOL DESIGNER Age 27. Technical education. Machine shop training, with experience in designing tools, jigs, fixtures, dies and special production equipment. Able to plan operations. Well versed in modern production methods. Open for engagement in shop or office. Pacific coast preferred.

0535 PRODUCTION MANAGER or shop superintendent with 8 years' experience in experimental, factory cost, planning and modern personnel work; also designing bombing planes, inspection, drafting and instruction. College education. Age, 30; married. New York or vicinity preferred but would consider other location.

0536 ENGINEER College graduate with 10 years' experience in tool design, transmissions, shop methods, drafting, inspection, factory layout and designing parts; also machine work on universal-joints. Good references supplied. Age, 28; married. Location immaterial.

(Continued on page 76)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE

Continued

MEN AVAILABLE

0537 MECHANICAL ENGINEER Technical graduate desires position with firm building automobiles or automobile engines. Is willing to start in a minor position with opportunity to be in contact with high-grade, aggressive men, and where, when worth is proved, advancement would follow to more responsible position. Age, 26; single. Pacific coast preferred but not essential. Available July 1.

0540 SALES ENGINEER possessing a wide acquaintance among the automotive engineers and purchasing agents in Michigan, Indiana, Ohio and Wisconsin desires position on either salary or commission basis. Specialty, pistons and piston-rings. Age, 35; married.

0541 MECHANICAL ENGINEER Technical graduate with 5 years' experience in the design and production of medium priced passenger cars with well-known company, also 3 years in sales and service work and 2 years in aeronautic work, desires position as engineer or assistant executive. Has considerable experience in technical writing and research. Age, 35; married. Mid-west location preferred but not essential.

0542 MECHANICAL ENGINEER well trained in scientific and engineering fields, with wide experience in the various branches of the industry, including production, inspection, testing and designing. Has made a special study of engine and chassis design and possesses a broad and well-balanced knowledge of the requirements necessary for bringing out a product that will compete successfully in the open markets and that will satisfy the demands of the buyer of tomorrow. Age, 35; married. Only a permanent connection considered. Address: G. A. L., Box 273, Conneautville, Pa.

0543 AUTOMOTIVE ENGINEER with 22 years' practical experience including technical training, designing, producing and marketing high-grade motor cars, trucks, busses and special body work desires to connect with firm planning to carry out new designs and who will appreciate conscientious worker. Executive engineer as well as competent correspondent with thorough knowledge of the material field and costs and capable of producing exclusive and original models. Can submit developed designs of originality and exceptional possibilities for the manufacturer. Available at once. Age, 41; married. Central West preferred but not essential.

0552 ASSISTANT CHIEF ENGINEER with 15 years' automotive experience in the largest automobile plants in the country and successfully holding the same position at present. Member S. A. E. for past 6 years.

0553 EXPERIMENTAL AND RESEARCH ENGINEER Recognized authority in this field who has held position of consulting engineer for two of the most prominent companies in the country. Age, 43; married. Eastern location preferred.

0554 AUTOMOTIVE ENGINEER Technical graduate with 12 years' practical experience in designing, production, system installation on trucks both gas and electric, bearings, batteries, and accessories. Has had a wide experience on production and three years in shop and industrial work. Fully capable to plan and carry out. Production work in the East preferred. Available at once. Married; age, 38.

0555 MECHANICAL ENGINEER Technical graduate with 7 years' practical engineering experience in experimental, design, physical testing and inspection; three years in automotive and 1½ years in ball bearing work. Age, 27; married. Best of references. Any location.

0556 CHIEF ENGINEER OR FACTORY MANAGER Eighteen years' experience in design, production, service and marketing of metal products. Technical graduate; 37 years old; married. Highest references as to ability, integrity and energy. Particularly familiar with accurately finished parts produced in large quantities. Any location. Available on short notice.

(*Continued on page* 78)

See announcement at the head of the S. A. E. Employment Service column, page 58.


BROCKWAY

S. A. E. EMPLOYMENT SERVICE
Continued

MEN AVAILABLE

0557 AUTOMOTIVE ENGINEER Technical graduate who is especially interested in gas-engine development. Experienced in experimental testing, designing, production, assembly, dynamometer test of eight-cylinder V-type light truck and tractor engines; 6 months with prominent company as machine operator, adjuster, toolsetter and inspector and one year with Government studying steam and gas powerplants. Available now. Any location. Age, 27; single.

0558 TECHNICAL GRADUATE, 30 years of age desires to make a change in location. Five and one-half years' experience as metallurgist and engineer of tests in ball-bearing industry and 1½ years as metallurgist in motorcycle industry. Any location. Married.

0559 ENGINEER with extensive experience in experimental field work, gas engineering, installing internal-combustion engines and general assembling; also research and inspection. Best of references furnished.

0560 AUTOMOTIVE ENGINEER who has had 6 years' experience as layout man and designer on general chassis work would like position in automotive or allied line where above experience can be used. Age, 26; married. New York City or vicinity preferred.

0561 GRADUATE ENGINEER Ex-service man with experience in manufacturing tools, gages, jigs and fixtures, also in production, inspection, magnetos and components and sales. Has held position of assistant inspection engineer. Age, 33; married.

0566 ENGINEER Carbureter specialist with 20 years' experience in all branches of automotive industry. Age, 48; married. Location immaterial.

0569 ENGINEER is available for a position. An energetic worker, producer and a close student of the business with extensive experience in sales work. Has held positions of branch and district manager. Married. Headquarters wherever desirable.

0570 ENGINEER with broad experience in the design of trucks and truck axles.

0574 ENGINEER with 5 years' experience in the automotive industry, desires a position with automobile, truck, tractor, engine or accessory company as manager, assistant manager, salesman or representative. Possesses executive ability and has held the positions of manager and purchasing agent. Would consider foreign location. Age, 26; single. Available at once.

0575 SALES AND AUTOMOTIVE ENGINEER is available as executive in sales or advertising, branch management, or export supervision. Has had 18 years' experience in automotive work including sales, advertising, manufacturing organization, service finance and exporting of automobiles, trucks and accessories. Now occupies position as New York branch manager for prominent automobile company but seeks greater opportunity and larger income. New York or Metropolitan District preferred. Age, 39. Available at an early date. References of highest character and record of personal achievement available.

0576 WRITER on automotive electrical subjects is available for either permanent position or assignment writing. Has written many electrical articles for one of the largest automobile trade papers in the country; also course on automobile ignition. Many years' experience in service, engineering, and teaching of automobile electrical subjects. Details on request.

0577 RADIO WRITER The author of many electrical articles for one of the largest automobile trade papers in the country and experienced in teaching and presenting difficult subjects in a simple way desires connection with a reliable publisher who wishes to put out a really good book on this subject. At present is writing book on Radio Receiving, which is neither a technical treatise nor a cheap edition hitting the high spots only, but a careful explanation of

(*Continued on page* 80)

See announcement at the head of the S. A. E. Employment Service column, page 58.

S. A. E. EMPLOYMENT SERVICE
Continued

MEN AVAILABLE

underlying principles and their practical application and is suitable not only for individuals but also for teaching the subject in high schools. Further information or samples of first chapters on request.

0578 MECHANICAL ENGINEER wants position with a progressive company. Fourteen years' practical experience in the design of trucks, tractors, engines and other work and in charge of engineering department. Also possesses shop experience on manufacturing methods and experiments on varied product. Age, 36; single. Location, immaterial.

0582 BODY ENGINEER Expert draftsman and body builder with full knowledge of the construction of all kinds of high-class custom body work as well as sample bodies, forms, jigs, etc., for production purposes. Is familiar with painting, trimming, writing of specifications and general manufacturing practice acquired through more than 20 years' experience in the vehicle trade.

0583 YOUNG ENGINEER with experience in airship and airplane engineering, research and production. Age, 24; single. Technical graduate.

0584 GRADUATE MECHANICAL ENGINEER with 8 years' experience in automotive design and production. Has held position as designer and chief engineer and is experienced also in experimental work, testing and production, of rotary valve vertical and V type engines. New England or New York City preferred. Salary reasonable. Available at once.

0585 ENGINEER Technical graduate with exceptional education and experience who has held positions of mechanical engineer in designing and experimental work on air-cooled and water-cooled six-cylinder cars, and consulting engineer both abroad and in this country. Experienced in internal-combustion engines, automotive products and special mechanical apparatus; instruction at officers training school in mechanical transport subjects and airplane engine design; advisory work as well as consulting work on automobiles and automotive products and purchasing. Best of references.

0586 MAN with 17 years' practical experience in automotive construction and competent to act as production manager or engineer or supervisor of inspection, would consider position as sales engineer or mechanical representative. At present employed in this capacity. Age, 41; married. Detroit preferred but not essential.

0589 CHIEF INSPECTOR desires to establish connection with a good company with chance for advancement in that capacity or as assistant engineer or chief draftsman. Fifteen years' experience in automobile engineering and production. Middle West preferred. Address, G. H. S., 99 Marston avenue, Detroit.

0590 EXECUTIVE AND DESIGNING ENGINEER Patentee of engines, axles, etc., with long years of experience with leading builders of automotive vehicles and their component units and parts will be available July 1. Location immaterial.

0591 ENGINEER Man with technical education and 4 years' experience in automotive design and construction who is capable of taking a position of responsibility with an automobile or aeronautical company. Age, 24; single. New York City or vicinity preferred. Available June 15.

0592 ENGINEER English university graduate who served for 5 years in Army as inspector of mechanical transport. Has held positions of marine engineer, designer of engines, tractors, electrical work, pumps, airplanes, carbureters and investigating liquid fuels and has been with a prominent company making special investigations on engine problems as technical advisor, production manager, designer, and development engineer. At present manufacturing radio apparatus but is available any time. No preference as to location. Would go abroad. Age, 44; married.

(*Continued on page* 82)

See announcement at the head of the S. A. E. Employment Service column, page 58.

ZENITH

S. A. E. Employment Service

Continued

MEN AVAILABLE

0593 ENGINEER with 4 years' experience as mechanical draftsman, one year as department supervisor and 2 years as sole distributor for states of Pennsylvania and New Jersey for firm dealing in automobile parts. Position of distributor preferred but not essential.

0594 ENGINEER Aged 33 and married is available for a position. Has held positions of superintendent of machine shop in charge of machine work on street cars, steam cars and buses; also has had charge of the manufacture of automobile front axles, rear axles and steering gears. Was chief inspector for 3 years with Government, foreman, toolmaker doing experimental work and manager of garage and machine shop for 3 years. Massachusetts preferred but not essential. Available now.

0595 SERVICE ENGINEER with wide experience in handling fleet equipment with executive ability in handling men and getting results. Graduate engineer with exceptional ability on heavy-duty and truck engines. Five years' experience in manufacturing and 9 years' service work. Reliable and competent. Location, Boston. Available in 10 weeks' time. Best of references supplied.

POSITIONS AVAILABLE

229 MECHANICAL RESEARCH ENGINEER with good technical education and well established in theory and experienced in practice, both in the design of the mechanism used in computing and typewriting machines together with a knowledge of tooling methods for the accurate production of small stamped and machined parts requiring close tolerances. A man preferably American born and about 35 years old who, within the past 3 or 4 years, has been employed in experimental development work. Must be willing to develop the ideas of others until he proves his ability to develop his own and be capable of managing a department composed of engineers and mechanics. Position offers a chance to advance to assistant chief engineer.

230 SALES MANAGER with practical uptodate methods for the distribution of passenger cars and trucks is wanted by a live new organization. Preferably a man familiar with approved distribution and sales discount methods. Location, Boston.

241 SALESMEN Two or three first-class men are wanted by a company in Philadelphia. Necessary to have unquestioned production records as city salesmen on automobiles in a retail way.

245 SALESMEN with sufficient engineering ability so that they are familiar with internal-combustion engine detail and can talk with repair shop men on subjects in which they are interested and in which the company is interested, particula[illegible]s, cylinders and the packing of pistons. Considerab[illegible] territory open for men having this ability and several branches open where sales managers are needed. Location, New York City.

*258 BODY LAYOUT AND DETAIL DRAFTSMAN A first-class man is wanted by plant located at Syracuse.

266 WHOLESALE AUTOMOBILE SALESMEN One or two good men to work under district manager, covering several states in Chicago territory.

*269 MAN to call upon the engineers and sales managers of the automobile builders. Must possess sales ability in connection with engineering experience, be able to talk with the automotive engineers in their own language, and have the ability to put sales over. Location, Albany.

270 SALESMEN and district sales managers are needed by company in New York City.

271 SALES MANAGER with mechanical knowledge and thorough sales experience to manage the sales of an automotive specialty. Unusual opportunity for an energetic ambitious man to build a future. Location, New York City.

(*Continued on page* 86)

See announcement at the head of the S. A. E. Employment Service column, page 58.

DURA

WINDOW REGULATORS

DURA Window Regulators are simple in design, easily operated, and the utmost in reliable efficiency.

The DURA COMPANY *Toledo, Ohio.*

DURA

WINDOW REGULATORS

We Sit on Your Side of the Desk

THERE are only about 129 automobile manufacturers in the United States. That means, in building our own business, we *had* to sit on *your* side of the desk.

Starting with only two customers in 1917, our business has grown until now Dura Window Regulators are used on far *more* closed cars than any other.

We are proud of having this reputation — *once a contract is given to the Dura Company, a manufacturer knows everything will be all right.* He knows he will get the easiest and simplest ventilation in his closed models, for the least final cost.

We start with your body drafts and lay in the best of the three types of that particular job. Then the samples are made and checked by a sample installation.

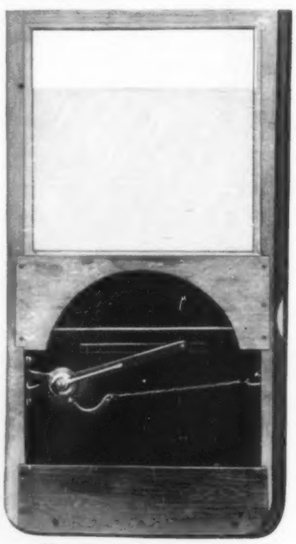
Our man is on the job with the arrival of the first quantity shipment. He shows all we know about installing regulators accurately and economically.

This installation service continues as often as necessary so long as production continues. All this we believe is good business insurance.

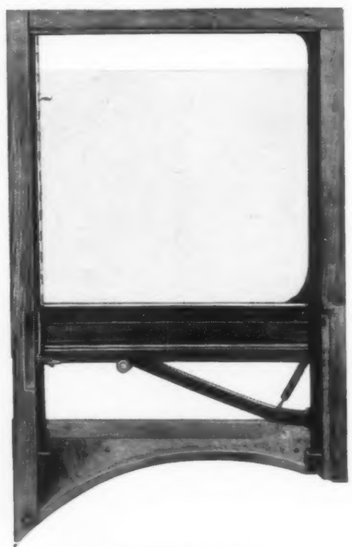
In the last analysis, we are both striving to please the ultimate consumer—the closed car owner!

For no matter how well we make our regulators, if they are not *properly installed* in the proper windows, they cannot render easy, quiet and *permanent* satisfaction to the car owner.

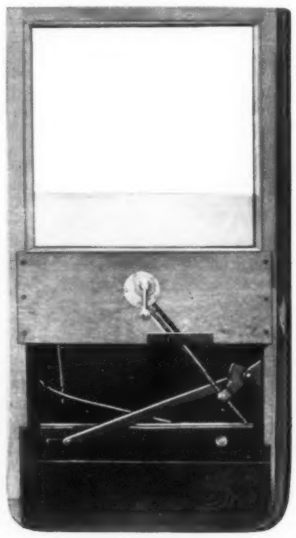
Let us consult with you concerning *your* needs.



LEVER TYPE
This quick-acting, Lever Type Dura raises or lowers the window instantly — with less than half a turn.



CLEMENT TYPE
This type most satisfactorily meets every need for rear windows or other windows where the operating handle must be out of the way.



CRANK TYPE
A crank operating regulator with less than thirty-five parts with the simplest locking means known.

The DURA COMPANY — *Toledo, Ohio.*

S. A. E. EMPLOYMENT SERVICE
Continued

POSITIONS AVAILABLE

*272 DIE-CASTING REPRESENTATIVE Strictly high-grade experienced man is desired by plant located in Pennsylvania.

*273 SALES ENGINEER for engine and transmission department. Should be a graduate of some good technical school as well as have had practical experience in the design and building of engines and transmissions; not an experimenter but one that can produce the goods is desired. Location, Pennsylvania.

275 AUTOMOTIVE ENGINEER Company wishes to get in touch with middle aged man of high reputation and ability with editorial and advertising experience whose general appearance will invite interview and confidence. An investment of several thousand dollars will be necessary, and large returns may be easily realized by investor. Full particulars at interview, but not by mail. Address, Miller & Glass, 289 Fourth Avenue, New York City.

280 SENIOR ENGINEER Salary $2,100-$2,700 and junior engineer, salary $1,320-$1,980 with Interstate Commerce Commission for the valuation of property of common carriers. For further information address John T. Doyle, secretary United States Civil Service Commission, City of Washington.

281 JUNIOR ENGINEER AND DECK OFFICER in the United States Coast and Geodetic Survey, salary $2,000-$2,240. Examination until June 22. For further information address John T. Doyle, secretary United States Civil Service Commission, City of Washington.

283 COMPUTER in the United States Coast and Geodetic Survey, for duty in the City of Washington, at a salary of $1,400 and at Manila, P. I., at a salary of $2,000. For further information, address John T. Doyle, secretary, United States Civil Service Commission, City of Washington. Examination until Aug. 10.

284 ASSISTANT EXAMINER in the Patent Office at Washington; salary $1,500-$3,900. Examination until Aug. 23. Further information can be secured from John T. Doyle, secretary, United States Civil Service Commission, city of Washington.

285 ENGINEER in the Bureau of Standards, City of Washington at a salary of $2,800-$4,000; also associate engineer at a salary of $2,000-$2,800 and assistant engineer with a salary of $1,500-$1,800. For further information address John T. Doyle, secretary, United States Civil Service Commission, City of Washington.

286 TECHNICAL SALESMAN Manufacturing corporation raising its own capital to market a most profitable line of new machinery, can use a few salesmen to aid in the present financing campaign. After sufficient capital is raised, the men will be placed in manufacturing sales department where large earnings are assured. Call at the office to see the machinery and investigate the proposition. Address: John Cetrule, secretary, Triplex Machine Tool Corporation, 18 East 41st Street, New York City.

*287 SALES ENGINEER with experience in bearing engineering, preferably a man with ball bearing engineering and sales experience and familiar with the automotive industry. Location, Philadelphia.

289 ENGINEER wanted for transportation department. Work involves making detailed analyses of delivery systems and will eventually lead to sales work. Among other things state age, previous experience and education. Location, suburb of New York City.

*291 CARBURETER SALES ENGINEER Carbureter manufacturer has an opening for a man who must have technical ability to make factory tests and be able to talk to factory engineers in their own language. This is a road job working out of

(*Continued on page* 88)

See announcement at the head of the S. A. E. Employment Service column, page 58.


Strom
BEARINGs
STUTZ
Strom Bearing Equipped
U. S. BALL BEARING MFG. COMPANY
(Conrad Patent Licensee)
4533 Palmer Street - CHICAGO, ILLINOIS

S. A. E. EMPLOYMENT SERVICE

Continued

POSITIONS AVAILABLE

Chicago, where the man will receive training on this particular carbureter. Straight salary and expenses while away from home.

*292 ENGINEER wanted to take charge of department as foreman. Should be a man of mature experience, particularly one who has had thorough experience in modern plants manufacturing machine tools or on fine machine work and capable of handling men. Location, Massachusetts.

*296 HIGHWAY TRANSPORT EXPERT Man who is thoroughly familiar with highway transport program, has good standing among educators, understands the value of publicity and can stimulate the compilation of data relating to the economic field of highway transport and secure its distribution to the colleges, high schools and other educational institutions of the country is wanted. Location, city of Washington.

297 SOLICITOR for cylinder grinding, pistons, rings, etc., is needed by company on Long Island. Must be at least 30 years old, own an automobile and be able to sell a profitable garage accessory as a side line. Splendid opportunity for right man. Salary and drawing account against substantial commissions. Territory, New York City and Long Island.

*298 YOUNG MAN with some experience as draftsman or designer is wanted by company in New York City. Prefer a man not long out of school who has had a little experience along this line.

299 SUPERINTENDENT for plant in eastern Pennsylvania manufacturing automobile axles. Write, giving all details as to qualifications and salary wanted, in first letter.

300 AIRPLANE ENGINE DESIGNER Man experienced in airplane work preferred, but an automotive man would do. Location, New Jersey.

301 GOOD TRACER AND DETAILER is wanted by New Jersey airplane company.

302 INSTRUCTOR on starting, lighting and ignition. Several evenings a week. Not a day school proposition. Location, Brooklyn.

303 AUTOMOTIVE SCHOOL PRINCIPAL Day school and some night school work included. Location, Brooklyn.

304 DRAFTSMEN AND DESIGNERS Several men who are experienced in automobile design and are willing to work on board and do various kinds of work. Location, New York City.

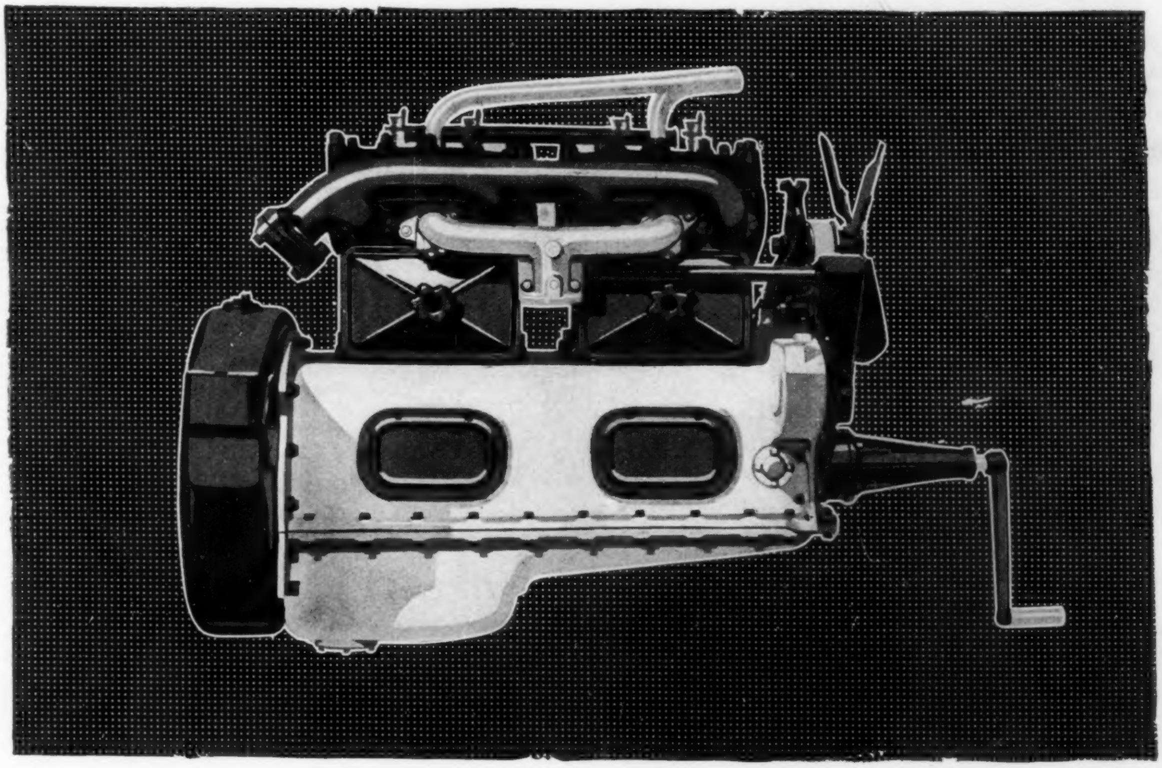
305 SERVICE Men wanted at factory of prominent motor-truck company in Michigan. Will receive special training as to methods and policies of doing business at the branches and after training is received, they will be placed in the various branches as service manager in which position they will have charge of the repair-shop, stockroom and the handling of customers along the lines of service. Maximum salaries at start between $225 and $250 per month. Should have mechanical knowledge, ability to lay out the work in the repair-shop, and executive ability along the lines of supervising the mechanics and the personnel of the department as well as proper personality for handling customers. If application is satisfactory, interview will be arranged.

306 BODY DESIGNER High-class man wanted by prominent company in Ohio.

307 MEN who are essentially manufacturers, are available in the automobile parts manufacturing business and possess broad enough experience to enable them to discuss mechanical problems arising in the trade and installation of such parts as axles and other car units are wanted by plant located in Ohio.

(*Concluded on page* 90)

See announcement at the head of the S. A. E. Employment Service column, page 58.



S. A. E. EMPLOYMENT SERVICE

Concluded

POSITIONS AVAILABLE

308 MECHANICAL AND PRODUCTION ENGINEER who is experienced on taper roller bearings, preferably man who has been employed by the Timken Roller Bearing Co., is wanted immediately. Must have wide experience in all details and will make his headquarters in New York City.

*309 ENGINEER Well trained man of ten or more years' experience on automotive engine design is desired by company in Middle West. State experience and salary in first letter.

310 ENGINEER wanted who is able to design jigs, do his own drawing and tracing, work around the plant, and make suggestions for better production methods and at same time take into consideration the advice of others who are as vitally concerned as he might be. This is a manufacturing business that consists mostly of steering connecting-rods, steering cross-tubes, hood fasteners, screw-machine products and robe and foot-rails. Location, Ohio.

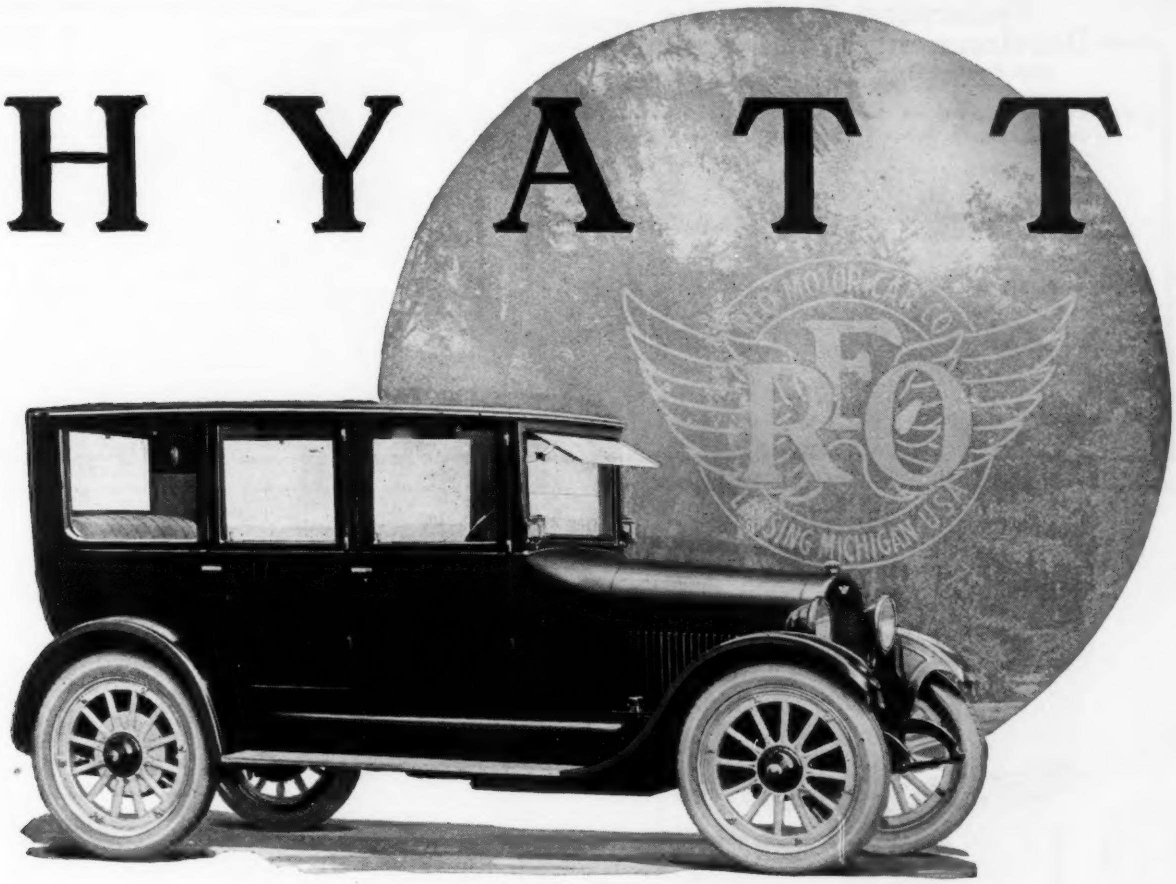
*311 MAN capable of acting as treasurer of automotive company, to take entire charge of finances. Must have had experience in automotive line. Salary $6,000 or more.

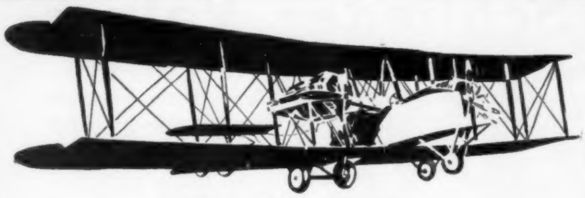
312 SALESMEN Several senior and several junior salesmen wanted for Metropolitan territory.

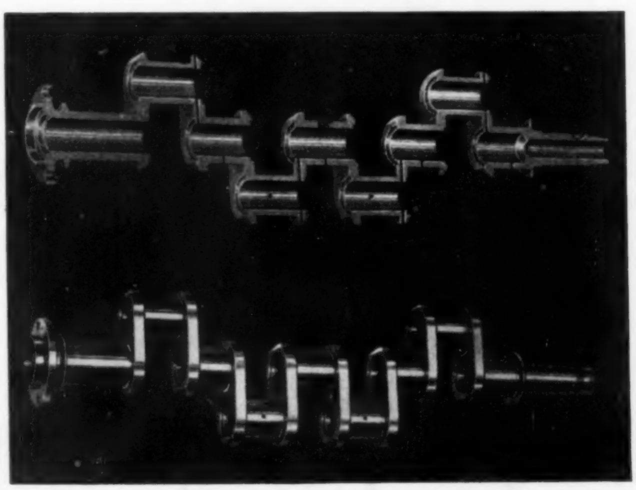
*313 DESIGNER AND DRAFTSMAN Experienced man for designing automobile engines. Location, Syracuse.

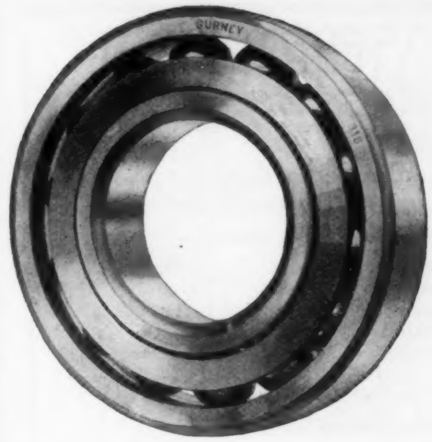
314 INSTRUCTOR is wanted for automobile classes and elementary branches of high-grade private institution in Hawaii. Duty starts in September. Address, L. C. Lyman, Hilo Boarding School, Hilo, T. H., or J. Leslie Putnam, Y. M. C. A. Central Department, Honolulu, T. H.

See announcement at the head of the S. A. E. Employment Service column, page 58.


REO







GURNEY


QUALITY SNAP RINGS
"Cast to Last"
Easy to Install
Quick Seating
Long Lived
Accurate
DEPENDABILITY REPUTATION
AUTOMOTIVE EQUIPMENT ASS'N
JUSTICE QUALITY
ROLLS-ROYCE
OF AMERICA, INC.
is *also* equipped
with Quality
Snap Rings. . . .
More Than a
Million a
Month
QUALITY
SNAP RINGS
12-4 1/2 .015 X 5/16
THE Piston RING COMPANY
MUSKEGON, MICH.

SOURCES OF SUPPLIES

An Index to Advertisers' Products Marketed as Conforming With S.A.E. Standards and Recommended Practices and to Products for Which There Are No S.A.E. Standards

(Continued on page 110)

EXPLANATION OF SYMBOLS

Parts and Materials followed by key numbers have been standardized by the S. A. E. The numbers refer to S. A. E. HANDBOOK data sheets on which each standard is published.

*Companies whose names are preceded by an asterisk supply the parts or materials under which the company is listed as conforming with the S. A. E. Standard referred to.

**Parts and Materials followed by two asterisks indicate that two or more S. A. E. Standards are applicable. Information as to standards incorporated should be obtained from the manufacturer.

The addresses of companies listed in Sources of Supplies can be obtained from their current advertisements indexed on page 114.

SOURCES OF SUPPLIES

Chains, Roller, E3
*Link-Belt Co.
*Whitney Mfg. Co.
Chains, Silent, E2
*Link-Belt Co.
*Morse Chain Co.
Whitney Mfg. Co.
Chains, Tire, Motor-Truck
Arrow Grip Mfg. Co., Inc.
Channels, Window Glass
Dahlstrom Metallic Door Co.
Clamps, Hose, C51
Schrader's Son, Inc., A.
Clamps, Machinists'
Eberhard Mfg. Co.
Williams, J. H., & Co.
Cleaners, Windshield
Folberth Auto Specialty Co.
Clutches, Engine, A1 and A2
*Brown-Lipe Gear Co.
Clutches, Power Transmission
Link-Belt Co.
Coils, Electrical Equipment
Acme Wire Co.
Conduit, Flexible, C52
*Titeflex Metal Hose Corporation
Connections, Tire-Pump
Schrader's Son, Inc., A.
Controls**
*Brown-Lipe Gear Co.
Cooling Systems**
*G & O Mfg. Co.
*Long Mfg. Co.
National Can Co.
Couplings, Flexible
Thermoid Rubber Co.
Cranes, Portable Electric
Link-Belt Co.
Crankshafts
Canton Drop Forging & Mfg. Co.
Jackson Motor Shaft Co.
Park Drop Forge Co.
Union Switch & Signal Co.
Vibration Specialty Co.
Wyman-Gordon Co.
Cups, Oil and Grease, C57
*Bowen Products Corporation
Cutters, Milling
Haynes Stellite Co.
Cutters, Woodruff
Whitney Mfg. Co.
Dampers, Vibration
Vibration Specialty Co.
Dashes
Parish Mfg. Corporation
Dies
Greenfield Tap & Die Corporation
Distributors, Ignition, B16
*Atwater Kent Mfg. Co.
*North East Electric Co.
Dogs
Williams & Co., J. H.
Door-Caps
Dahlstrom Metallic Door Co.
Drills, Twist
Greenfield Tap & Die Corporation
Drive, Starting Motor
Eclipse Machine Co.
Drop-Forgings
Brewer-Titchener Corporation
Canton Drop Forging & Mfg. Co.
Champion Machine & Forging Co.
Park Drop Forge Co.
Sheldon Axle & Spring Co.
Spicer Mfg. Corporation
Union Switch & Signal Co.
Universal Machine Co.
Williams & Co., J. H.
Wyman-Gordon Co.
Ends, Tank-Strap
Brewer-Titchener Corporation
Engines, Motor Truck**
*Continental Motors Corporation
*Light Mfg. & Foundry Co.
*Waukesha Motor Co.
Engines, Passenger Car**
*Continental Motors Corporation
*Light Mfg. & Foundry Co.
Engines, Steam, Automotive
Stumpf Una-Flow Engine Co., Inc.
Engines, Tractor**
*Continental Motors Corporation
*Light Mfg. & Foundry Co.
*Waukesha Motor Co.
Facings, Clutch
Johns-Manville, Inc.
Thermoid Rubber Co.
Fans
Service Products Corporation
Fans, Radiator, A14a
*Detroit Carrier & Mfg. Co.
Sparks-Withington Co.
Fasteners, Hood
Eberhard Mfg. Co.
Felt
American Felt Co.

Felloe-Bands, Motor-Truck, Pneumatic Tire, G4
*Motor Wheel Corporation
Fender, Irons
Brewer-Titchener Corporation
Fenders
Parish Mfg. Corporation
Fibre, Vulcanized
American Vulcanized Fibre Co.
Flanges, Hub
Crosby Co.
Parish & Bingham Corporation
Smith Corporation, A. O.
Floors, Metal
Dahlstrom Metallic Door Co.
Flow-Meters
Penberthy Injector Co.
Followers, Cam
Wilcox Motor Parts & Mfg. Co.
Forgings, Brass
Scovill Mfg. Co.
Forgings, Drop (See Drop-Forgings)
Frames, Pressed Steel
Parish & Bingham Corporation
Parish Mfg. Corporation
Smith Corporation, A. O.
Frames, Rolled Section
Mansfield Steel Corporation
Fuel-Feed Systems
Sparks-Withington Co.
Fuses, Electric, B32
*Johns-Manville, Inc.
Gages, Gasoline
Kollmorgen Optical Corporation
Nagel Electric Co., W. G.
Penberthy Injector Co.
Gages, Limit
Greenfield Tap & Die Corporation
Gages, Oil, B12
*Nagel Electric Co., W. G.
Gages, Plain
Greenfield Tap & Die Corporation
Gages, Screw Thread
Greenfield Tap & Die Corporation
Gages, Tire Pressure
Schrader's Son, Inc., A.
Gasoline
Texas Co.
Gears, Bevel
Link-Belt Co.
Gears, Reduction
Waukesha Motor Co.
Gears, Spur
Link-Belt Co.
Gears, Transmission
Canton Drop Forging & Mfg. Co.
Link-Belt Co.
Wyman-Gordon Co.
Gears, Worm
Link-Belt Co.
Generators, Battery Charging (Electric Vehicle)
Westinghouse Electric & Mfg. Co.
Generators (Bracket Mounting, B18)
*Atwater Kent Mfg. Co.
*North East Electric Co.
Westinghouse Electric & Mfg. Co.
Generators (Flange Mounting, B17)
*Atwater Kent Mfg. Co.
*North East Electric Co.
Westinghouse Electric & Mfg. Co.
Glasses, Lamp, B6
*Smith Glass Co., L. E.
Glass, Windshield
Ainsworth Mfg. Co.
Graphite
Dixon Crucible Co., Jos.
Greases, Cup and Gear
Dixon Crucible Co., Jos.
Texas Co.
Grease-Cups, C57
*Link-Belt Co.
Guards, Radiator
Mansfield Steel Corporation
Handles, Control-Lever Ball, J2
*American Insulator Corporation.
Handles, Machine-Tool
Williams & Co., J. H.
Hardware, Body
Brewer-Titchener Corporation
Heaters, Passenger-Car Body
Perfection Heater & Mfg. Co.
Hinges, Door
Eberhard Mfg. Co.
Hinges, Windshield
Eberhard Mfg. Co.
Hitches, Trailer, K42
*Mansfield Steel Corporation
Hoists, Power and Hand
Mansfield Steel Corporation
Holders, Tool
Williams & Co., J. H.
Hooks, Towing
Mansfield Steel Corporation

Horns, Hand
E. A. Laboratories, Inc.
Horns, Motor-Driven
E. A. Laboratories, Inc.
North East Electric Co.
Sparks-Withington Co.
Hose, Gasoline
Thermoid Rubber Co.
Hose, Radiator, C51
*Thermoid Rubber Co.
Housings, Axle
Crosby Co.
Dixon Crucible Co., Jos.
Parish & Bingham Corporation
Parish Mfg. Corporation
Smith Corporation, A. O.
Hubs, Wheel
Salisbury Axle Co.
Smith Corporation, A. O.
Ignition-Generators
North East Electric Co.
Inserts, Spring
Center-Fed Spring Insert Sales Co.
Instruments, Vibration, Indicating and Recording
Vibration Specialty Co.
Insulation, Molded
General Bakelite Co.
Jacks, Wheel
Arrow Grip Mfg. Co., Inc.
Jars, Storage Battery, B39
*Marko Storage Battery Co.
Keys, Woodruff
Whitney Mfg. Co.
Keys, Spring
Williams & Co., J. H.
Lenses (See Glasses, Lamp)
Lining, Brake, C55
*Johns-Manville, Inc.
*Thermoid Rubber Co.
Lubricants, D138
Dixon Crucible Co., Jos.
Texas Co.
Lubricating Systems
Bassick Mfg. Co.
Bowen Products Corporation
Chassis Lubricating Corporation
Lubricators, Leaf Spring
Anderson Spring Lubricator Co.
Machines, Balancing
Olsen Testing Machine Co., Tinius
Vibration Specialty Co.
Machines, Bolt Threading
Universal Machine Co.
Machines, Grinding
Greenfield Tap & Die Corporation
Machines, Power Transmission
Link-Belt Co.
Machines, Testing
Olsen Testing Machine Co., Tinius
Magnetos, B14
*Bosch Magneto Co., Inc., Robert
*Scintilla Magneto Co., Inc.
*Teagle Co.
Moldings, Cold-Drawn Steel
Dahlstrom Metallic Door Co.
Molybdenum, Metallic
Vanadium Corporation of America
Motors, Electric Vehicle
Westinghouse Electric & Mfg. Co.
Motors (See Engines)
Motors, Starting (See Starting Motors)
Nails
Interstate Iron & Steel Co.
Nipples, Motorcycle, F4
*Budd Wheel Co.
Nuts, Thumb
Williams & Co., J. H.
Odometers, Hub, F2
*Johns-Manville Co., Inc.
Oils (See Lubricants)
Packing, Asbestos and Fibrous
Johns-Manville, Inc.
Pads, Accelerator
Auto Pedal Pad Co.
Pads, Pedal
Auto Pedal Pad Co.
Paints
Dixon Crucible Co., Jos.
Panels, Instrument
Dahlstrom Metallic Door Co.
Parts, Pressed Steel (See Stampings)
Patterns, Wood and Metal
Standard Pattern Works of Detroit Inc.

Pedals, Extension
Auto Pedal Pad Co.
Pencils, Drawing
American Lead Pencil Co.
Dixon Crucible Co., Jos.
Pins, Cotter, C7
*Williams & Co., J. H.
Pins, Piston
Universal Machine Co.
Wilcox Motor Parts & Mfg. Co.
Pipe Fittings, Compression Type, Fuel and Lubrication
Ohio Metal Products Co.
Piston-Rings
Piston Ring Co.
Wilcox Motor Parts & Mfg. Co.
Power Plants Industrial
Waukesha Motor Co.
Power Take-Offs, E1
Brown-Lipe Gear Co.
Prints, Photo Reproduction
Baker Garland Reproduction Co., Inc.
Products, Screw-Machine
Link-Belt Co.
Ohio Metal Products Co.
Rich Tool Co.
Scovill Mfg. Co.
Spicer Mfg. Corporation
Propeller-Shafts
Salisbury Axle Co.
Spicer Mfg. Corporation
Push-Rods
Wilcox Motor Parts & Mfg. Co.
Radiators**
*G & O Mfg. Co.
*Long Mfg. Co.
National Can Co.
Sparks-Withington Co.
Reamers
Greenfield Tap & Die Corporation
Re-atomizers, Gasoline
Penberthy Injector Co.
Rectifiers, Battery Charging
Westinghouse Electric & Mfg. Co.
Relays, Cut-Out
North East Electric Co.
Retainers, Ball
Bearings Co. of America
Ribbons, Fender
Dahlstrom Metallic Door Co.
Rims, Pneumatic Tire, G1 and G2
*Budd Wheel Co.
*Motor Wheel Corporation
Rivets, Steel
Interstate Iron & Steel Co.
Rivets, Brass and Copper
Rome Brass & Copper Co.
Scovill Mfg. Co.
Rod-Ends, C8
*Brewer-Titchener Corporation
*Eberhard Mfg. Co.
Rods, Brass, D114 and D115
*Rome Brass & Copper Co.
*Scovill Mfg. Co.
Rods, Bronze and Copper
Rome Brass & Copper Co.
Rods, Door Curtain
Brewer-Titchener Corporation
Rods, Fibre
American Vulcanized Fibre Co.
Roller Bearings (See Bearings, Roller)
Running-Boards
Parish & Bingham Corporation
Parish Mfg. Corporation
Smith Corporation, A. O.
Screws, Cap, C1
*Scovill Mfg. Co.
Screws, Thumb
Williams & Co., J. H.
Shaft-Ends, C14
*Brown-Lipe Gear Co.
Shafts, Rear Axle
Salisbury Axle Co.
Shapes (Extruded), Brass, Bronze and Copper
Rome Brass & Copper Co.
Shapes (Rolled), Brass, Bronze and Copper
Rome Brass & Copper Co.
Sheets, Brass, D111
*Scovill Mfg. Co.
Sheets, Brass, Bronze and Copper
Rome Brass & Copper Co.
Sheets, Fibre
American Vulcanized Fibre Co.
Shims, Laminated
Laminated Shim Co., Inc.
Sills, Body
Smith Corporation, A. O.

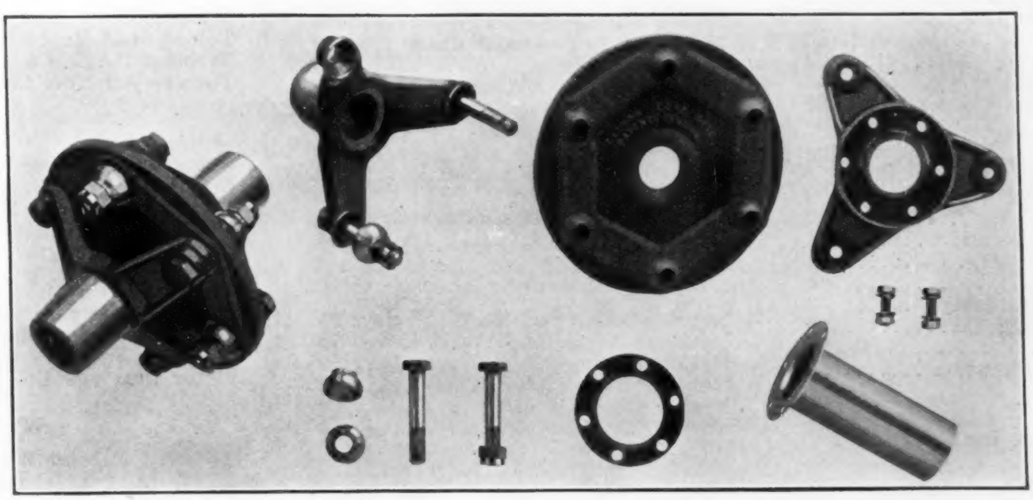
(Concluded on page 112)

The Index to Advertisers is given on page 114

SOURCES OF SUPPLIES

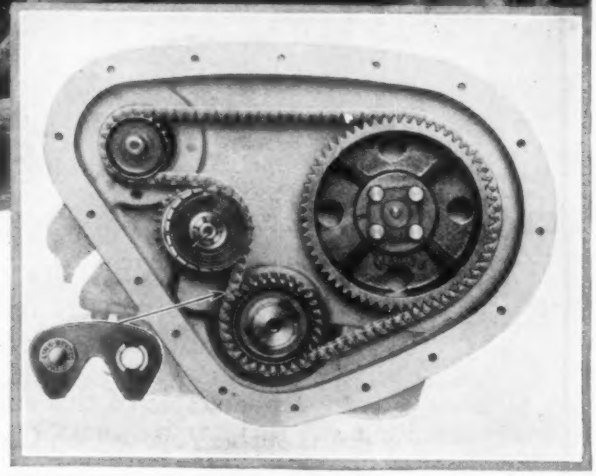
Sockets, Bow
Brewer-Titchener Corporation
Solder
Hoyt Metal Co.
Spokes, Motorcycle, F4
*Budd Wheel Co.
Spokes, Wood, Motor Truck, F1
*Motor Wheel Corporation
Spokes, Wood, Passenger Car, F1a
*Motor Wheel Corporation
Springs, Coiled
Gibson Co., Wm. D.
Springs, Flat
Gibson Co., Wm. D.
Springs, Leaf**
Sheldon Axle & Spring Co.
Springs, Motor Truck, H9
*Perfection Spring Co.
Standard Steel Spring Co.
Springs Passenger-Car, H8
*Perfection Spring Co.
Standard Steel Spring Co.
Sprockets, Roller-Chain, E4
*Link-Belt Co.
*Whitney Mfg. Co.
Sprockets, Silent-Chain
Link-Belt Co.
Whitney Mfg. Co.
Stabilizers
Watson Co., John Warren
Stampings
Brewer-Titchener Corporation
Crosby Co.
Motor Wheel Corporation
Parish & Bingham Corporation
Parish Mfg. Corporation
Smith Corporation, A. O.
Spicer Mfg. Corporation
Starter-Generators
North East Electric Co.
Starting Motors (Flange Mounting), B19
*Atwater Kent Mfg. Co.
*North East Electric Co.
Westinghouse Electric & Mfg. Co.
Starting Motors (Sleeve Mounting), B20
*Atwater Kent Mfg. Co.
*North East Electric Co.
Westinghouse Electric & Mfg. Co.
Starting, Automatic Meshing Drive for
Eclipse Machine Co.
Steel, Chromium, D6
*Central Steel Co.
*Interstate Iron & Steel Co.
Steel, Chromium - Vanadium, D6
*Central Steel Co.
*Interstate Iron & Steel Co.
Steel, Molybdenum
Interstate Iron & Steel Co.
Steel, Nickel, D5
*Central Steel Co.
*Interstate Iron & Steel Co.
*Union Drawn Steel Co.
Steel, Nickel-Chromium, D5
*Central Steel Co.
*Interstate Iron & Steel Co.
*Union Drawn Steel Co.
Steel, Silico-Manganese, D6
*Central Steel Co.
*Interstate Iron & Steel Co.
Steels, Carbon, D4
*Interstate Iron & Steel Co.
*Union Drawn Steel Co.
Steel, Screw Stock
Union Drawn Steel Co.
Steel, Spring
Interstate Iron & Steel Co.
Strips, Brass, Bronze and Copper
Rome Brass & Copper Co.
Superheat System, Gasoline
Deppé Motors Corporation
Switches, Lighting
Westinghouse Electric & Mfg. Co.
Switches, Starting
North East Electric Co.
Westinghouse Electric & Mfg. Co.
Systems, Lubrication (See Lubricating Systems)
Tacks
Interstate Iron & Steel Co.
Tachometers (with Standard Drive), C75
*Johns-Manville, Inc.
Tape, Insulated
Johns-Manville, Inc.
Tappets
Wilcox Motor Parts & Mfg. Co.
Taps
Greenfield Tap & Die Corporation
Terminals, Generator Cable, B22
American Insulator Corporation
Terminals, Starting Motor Cable, B21
American Insulator Corporation
Tire Carriers
Detroit Carrier & Mfg. Co.
Tire Locks
Detroit Carrier & Mfg. Co.
Tire-Pumps, Transmission Type, E1
Detroit Carrier & Mfg. Co.
Tires, Pneumatic, G1
Goodrich Rubber Co., B. F.
Goodyear Tire & Rubber Co.
Kelly-Springfield Tire Co.
*Thermoid Rubber Co.
Tires, Solid, G11
Kelly-Springfield Tire Co.
Tools
Haynes Stellite Co.
Williams & Co., J. H.
Top, Irons
Brewer-Titchener Corporation
Torque-Arms
Smith Corporation, A. O.
Tracings, Dupliacte
Baker Garland Reproduction Co., Inc.
Tractors, Industrial
Baker R & L Co.
Transmissions**
*Brown-Lipe Gear Co.
*Durston Gear Corporation
*Light Mfg. & Foundry Co.
Trucks, Industrial
Baker R & L Co.
Tubing, Brass, D116
*Rome Brass & Copper Co.
*Scovill Mfg. Co.
Tubing, Copper, D117
*Rome Brass & Copper Co.
*Scovill Mfg. Co.
Tubing (Tapered), Brass Bronze and Copper
Rome Brass & Copper Co.
Tubing, Fibre
American Vulcanized Fibre Co.
Tubing, Flexible Metal
Titeflex Metal Hose Corporation
Tubing, Rubber
Thermoid Rubber Co.
Tubing, Steel, D75
*Smith Corporation, A. O.
Tubing, Windshield
Dahlstrom Metallic Door Co.
Tungsten, Metallic
Vanadium Corporation of America
Univeral-Joints, E6
*Spicer Mfg. Corporation
*Thermoid Rubber Co.
Universal Machine Co.
Valve-Lifters
Rich Tool Co.
Valves, Fuel Reserve
Service Products Corporation
Valves, Poppet, A4
*Rich Tool Co.
Wilcox Motor Parts & Mfg. Co.
Valves, Tire
Schrader's Son, Inc., A.
Vaporizers (See Primers)
Ventilators, Cowl
Service Products Corporation
Voltmeters, B12
*Sterling Mfg. Co.
Westinghouse Electric & Mfg. Co.
Washers, Fibre
American Vulcanized Fibre Co.
Washers, Steel
Interstate Iron & Steel Co.
Wheels, Motorcycle, F3
*Budd Wheel Co.
Wheels, Pressed Steel Disc
*Budd Wheel Co.
Motor Wheel Corporation
Wheels, Wire, F1b
*Budd Wheel Co.
Wheels, Wood
Motor Wheel Corporation
Muncie Wheel Co.
Wicks, Felt
American Felt Co.
Windshields
Ainsworth Mfg. Co.
Wire (See Cable, Insulated)
Windows, Curtain
Brewer-Titchener Corporation
Wire, Magnet
Acme Wire Co.
Wire Products
Interstate Iron & Steel Co.
Wrenches
Greenfield Tap & Die Corporation
Williams & Co., J. H.

The Index to Advertisers is given on page 114



INDEX TO ADVERTISERS

An index to advertisers' products is given on pages 108, 110 and 112



"Howdy" Wilcox

Alton G. Seiberling